THE DESIGN OF A ROTATING CYLINDER HEAD VARIABLE COMPRESSION INTERNAL COMBUSTION ENGINE

A THESIS

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Master of Science in Mechanical Engineering

by

Thomas Benjamin Lane

June 1956

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Thomas B. Lane

THE DESIGN OF A ROTATING CYLINDER HEAD

VARIABLE COMPRESSION INTERNAL COMBUSTION ENGINE

Approved: ____________________

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Date approved by Chairman: 1 June 1956

to my future wife —

Ann Zanier Downs

(syla)

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I wish to express my sincere gratitude to all those members of the faculty and staff of the Georgia Institute of Technology who have contributed to the preparation of this investigation. Special appreciation goes to my thesis committee: Professor Robert L. Allen, thesis adviser, Dr. Ray L. Sweigert, Dean of the Graduate School, and Dr. Joseph P. Vidosic, who also served as my faculty adviser. I also wish to acknowledge Dr. Karl M. Murphy's contributions to the presentation of the paper.

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SUMMARY

The purpose of this study was the theoretical analysis and design of a rotating cylinder head valve arrangement which can be adjusted up or down in an engine cylinder so as to provide both the proper "breathing" facilities and a variable compression ratio.

The theoretical investigation of variable compression was conducted utilizing the computed combustion charts of Hershey, Eberhardt and Hottel. Otto engine cycles were calculated for compression ratios of 6 1/4, 7, 9 3/4, and 15:1 over a range of mixture inlet pressures from 4 to 14.7 pounds per square inch absolute. Appropriate curves were drawn and a correlation made between existing engines and the computed cycles. The Cadillac "Eldorado" engine was selected as a standard for comparison and efficiency increases in the order of 13 per cent were predicted possible over most of the operating range while employing a 15:1 compression ratio. Evidence indicated that cheaper fuels might also be used and low speed torque improved by over 10 per cent if the appropriate compression ratios were available to the "Eldorado" when needed. Reductions in operating costs of over 30 per cent were predicted possible for the variable compression Cadillac engine over the range investigated.

Based on a review of the past work of various experimenters, the rotary valve mechanism offers a number of inherent advantages over the conventional poppet valve system -- most notably, improvements in terms of "mechanical octanes" and "breathing" characteristics. Rotary valves

and variable compression appear compatible, and a valve system was devised which makes this combination possible. The problems of mechanisms, dimensions, sealing, lubrication, materials, balance and vibration are treated in this design as applied to a single cylinder 27.8 cubic inch displacement laboratory-type test engine.

Although conclusive results are not available from actual tests of the engine, some difficulties are anticipated, particularly with regard to lubrication and port sealing, and a suggested guide for development is provided.

It is felt that the rotary valve, variable compression engine offers great potentiality in providing all purpose performance including good low speed torque, reduced noise and vibration, exceptionally high power, improved thermal efficiencies, and reduced fuel costs. There must certainly be justification for research on this type of engine.

CHAPTER I

INTRODUCTION

Current trends in the development of internal combustion engines have centered about increasing their power and efficiency. Increases in compression ratio, reductions in friction and improvements in "breathing" have characterized the bulk of progress to date.

The purpose of this study was the theoretical analysis and design of a rotating cylinder head valve arrangement which can be adjusted up or down in an engine cylinder so as to provide both the proper "breathing" facilities and a variable compression ratio. To verify the design which grows out of the analysis will necessitate experimental testing of an actual engine using the valve as designed. Such testing will undoubtedly point to additional problems to be solved, particularly of a mechanical nature.* Since the engine is for developmental purposes, fabrication simplifications and arrangements for ready access are stressed.

The "rotary valve" engine is almost as old as Nikolaus Otto's invention of 1876 which was formulated on the theory originated by Beau de Rochas in 1862 (1).** F. W. Crossley experimented with rotary valves during the years between 1886 and the turn of the century; however,

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*Recommendations for future development of the engine valve design, along with suggestions of possible problems to overcome, are treated briefly in Appendix A.

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CHAPTER I

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by 1902, the contemporary poppet-valve engine had established such a record of reliability that the more cumbersome "sliding type" valve was almost completely rejected (2). Since that time, experimental work has progressed slowly, following favorable developments in materials and fabrication techniques. Among the outstanding pioneers in the field were C. W. Padget, Dugald Clerk, Burt McCollum, and C. V. Knight (3). Since World War I, very creditable work has been done by R. C. Cross (4) and F. M. Aspin (5). There were many others also working on this engine (6). At least one German engineer is known to be experimenting with ceramic rotors at present (7), and M. A. Zimmerman of Cleveland, Ohio is continuing the work of R. C. Cross on a horizontal rotary valve (8).

Although the theory of variable compression engines has long been known, as compared to the rotary valve engine a less significant amount of work has been done on the former, and no satisfactory unit is known to exist which utilizes the maximum potentialities of variable compression. The majority of the systems have employed mechanical linkages which were bulky and inadequate to meet constantly changing demands such as might be required of a typical automotive engine (9,10,11,12,13,14).

In the work which follows, an attempt is made to combine the favorable features of two principles -- varying compression and rotary valving -- into a single highly flexible, powerful and economical power plant.

CHAPTER II

THEORETICAL AND PRACTICAL INVESTIGATION OF PROPOSED ENGINE

<u>Variable Compression</u>.--A series of theoretical engine cycles is presented in support of incorporating variable compression in the proposed engine design. The cycles are computed for one pound of air reacting with a stipulated amount of fuel to give the usual combustion products. The chemical dissociation of the combustion products under high temperatures has been taken into consideration as suggested in the works of Hershey, Eberhardt, and Hottel (15).

A considerable amount of experimental work has been done in an effort to account for variations between actual operating engine cycles and those computed by the methods of Professor Hottel and co-workers (16, 17, 18). Withrow and Cornelius of the Research Laboratories Division, General Motors Corporation, have conducted a noteworthy analysis in which calculations indicate an average error of some nineteen per cent in the chart cycles (19); however, operating conditions were by no means typical in their laboratory equipment. An L-head engine was employed at an operating speed of 900 rpm. Compression ratios were 4.8:1 and 4.95:1.

Sample curves for computed values of indicated mean effective cylinder pressure (IMEP), thermal efficiency, volumetric efficiency, and maximum cylinder pressure (P_3) are plotted against variations in intake (manifold) pressure (P_1) in Figures 2, 3, and 4. Compression ratios are arbitrarily taken as 6 1/4:1, 7:1, 9 3/4:1, and 15:1.* The principal points of the throttled cycle are illustrated in Figure 1.

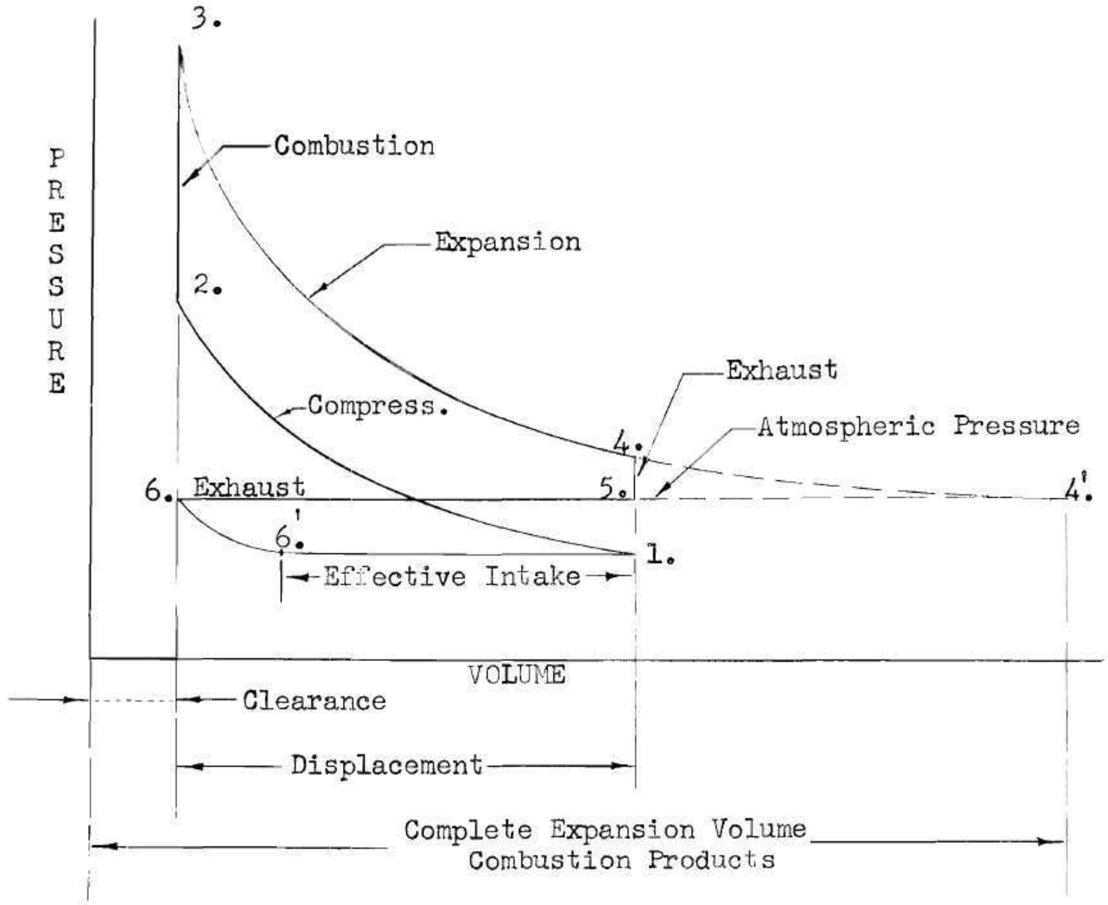

Figure 1. Assumed Combustion Cycle.

This type of computation is conventional procedure in actual laboratory engine analysis (20); however, variations due to gas friction in the ports and combustion chamber, heat transfer between the engine and mixture, and time delays in combustion are neglected in the

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*Refer to Appendix A, Tables 3, 4, 5 and 6.

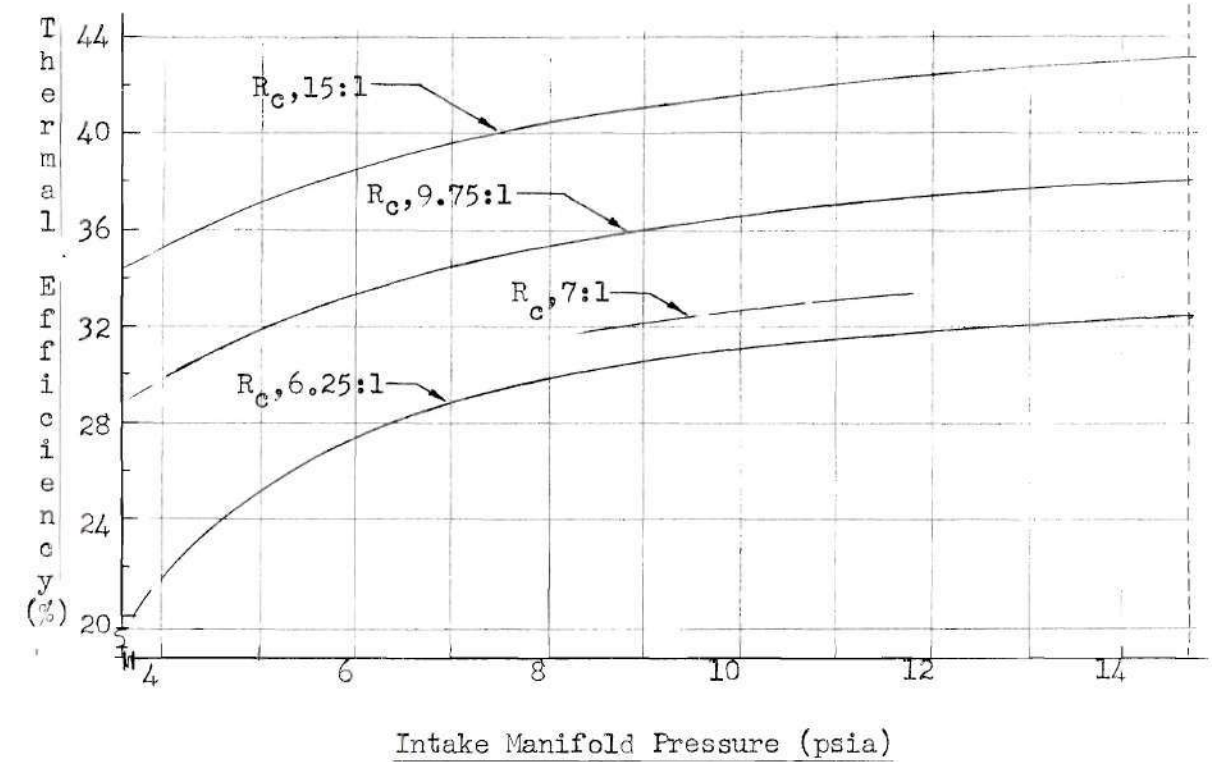

Figure 2. Computed Thermal Efficiency Curves

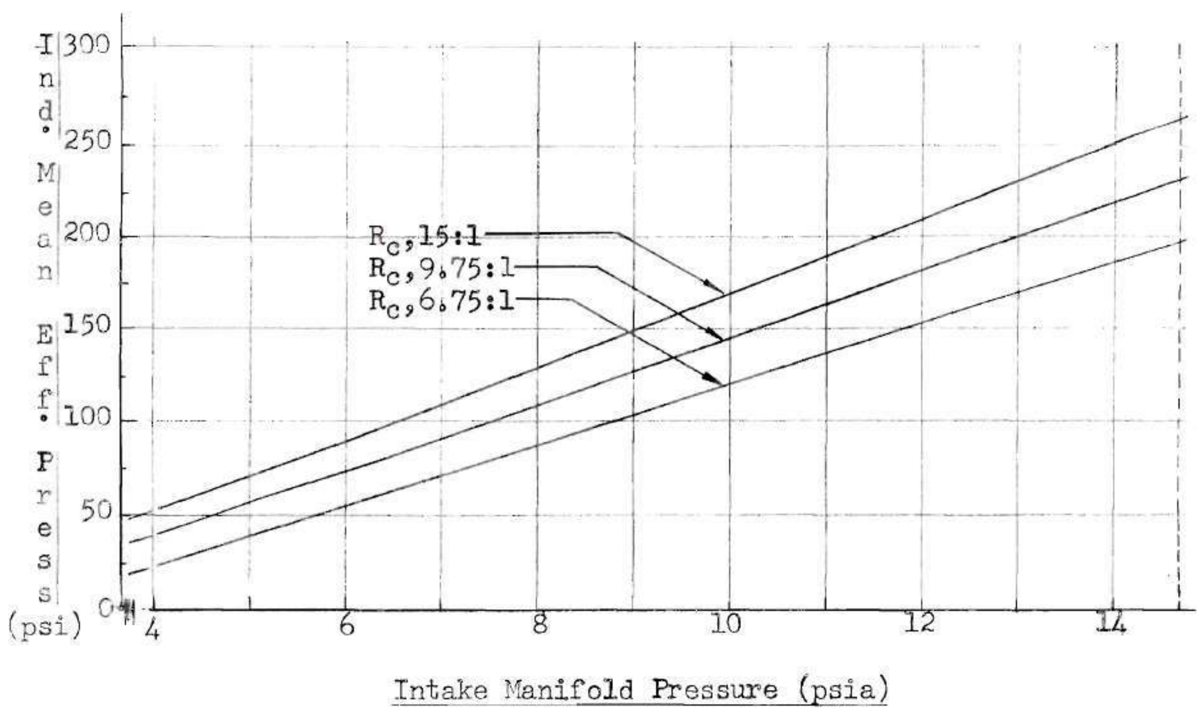

Figure 3. Computed Indicated Mean Effective Pressure (IMEP) Curves

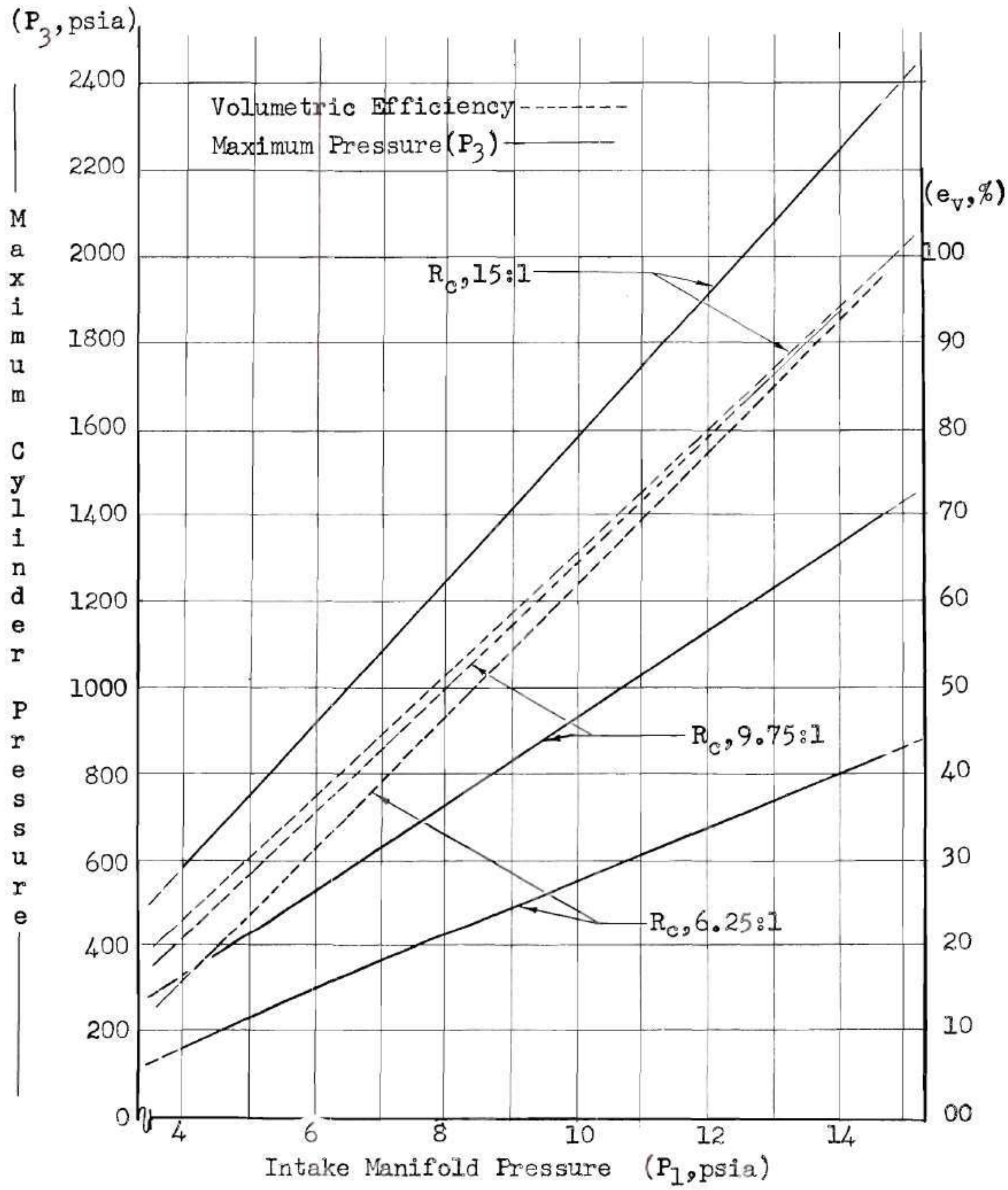

Figure 4.

Computed Maximum Cylinder Pressure
and
Volumetric Efficiency
Curves

computations. The rich mixture was used throughout in order to obtain better correlations with available full power ratings of the Cadillac engine indicated in Figure 5. Available information and data indicate relatively good breathing characteristics, and the design shows relatively compact combustion chambers in all engines under study;* hence, errors in the predicted values will tend to be minimized and the specified combustion charts should provide a satisfactory guide to comparison of the various engines.

A tabulation of operating characteristics for current model automobile, aircraft, and test engines is shown in Appendix A, Tables 7(a) and 7(b). Only overhead valve engines are included, since comparison of data from engines employing other valve arrangements has conclusively established that the majority of higher performance units is in this category (21, 22, 23, 24, 25, 26, 27).

Study of Table 7(a) indicates the current trend of automotive designs toward higher compression ratios in their effort for more economy and higher power. In order to more clearly understand the feasibility of these changes in the light of a complete absence of recent major fuel improvements, attention must be called to the large angles of valve overlap and the high engine speeds at which maximum torque is developed. The late closing of the inlet valve allows part of the charge to be expelled before compression at low speeds; thereby helping protect the engine

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*Appendix A, Table 8 indicates volumetric efficiencies up to seventy-eight per cent for the Cadillac "Eldorado".

from detonation due to excessive pressures from the firing of large charges of mixture. As speed is increased, volumetric efficiency improves correspondingly, and since higher engine speeds reduce engine "knock" tendencies (28), the greater charges can be used effectively by the engine. An excellent example of large valve overlap may be found in the 1956 Cadillac "Eldorado" engine. If a mechanical efficiency (η) of 90.7 per cent is assumed based on Callender's formula: $\eta = 0.98(1-\frac{0.3}{D})$ (where "D" is cylinder diameter in inches)(29), it is possible to calculate the brake mean effective pressure (BMEP) and thence the predicted indicated mean effective pressure (IMEP).* From this tabulation it is noted that a maximum indicated mean effective pressure of 182 psi is obtained at 3200 rpm and 9.75:1 compression ratio. From Figure 3, an intake pressure (P_1) of 12.2 psia corresponds to this indicated mean effective pressure and from Figure 4, the volumetric efficiency should reach 78 per cent.

From the assumption that pressure drop across the inlet valve at closing should approach zero for low engine speeds, it should be seen by the simple geometry of the system that a valve closing 105° after bottom center can capture a maximum of 43.5 per cent of the total displaced volume when heat transfer to the mixture is zero. A volumetric efficiency of 43.5 per cent corresponds to 97 psi ind. mean effective pressure. This obvious dissimilarity of values verifies the observation

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*Refer to Appendix A, Table 8 for tabulation of computations pertinent to the Cadillac engine. See Figure 5 for Cadillac test curves.

regarding increasing volumetric efficiency with speed (up to a maximum in the vicinity of maximum torque).

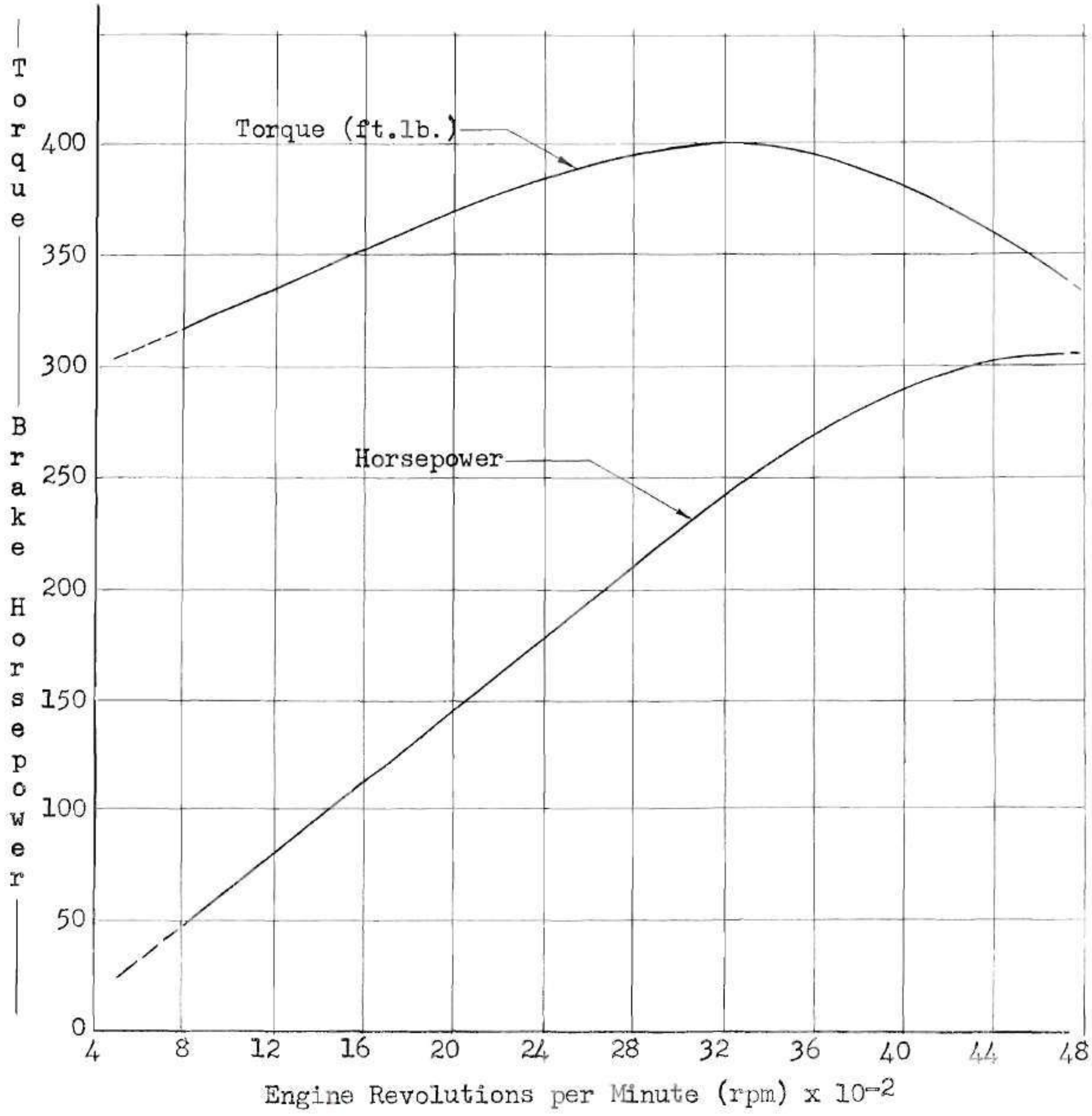

Figure 5. Performance Curves For 1956 Cadillac "Eldorado" (30)

Laboratory test data, corrected to S. A. E. Test Code.
Temperature air: 60°F; Atmospheric pressure: 29.92 in. Hg.

The curves for compression ratio 6 1/4:1 were computed so that comparison could be made with actual tests made on a 1948 model Chevrolet six-cylinder, overhead valve automobile engine installed in the mechanical engineering laboratory at the Georgia Institute of Technology.*

The following similarity of data is observed, based on the matching of indicated mean effective pressures between actual data, Table 8 and calculated curves, Figures 2 and 3:

1. When the estimated indicated mean effective pressure is 103 psi, the corresponding intake pressure of 9 psia (Figure 3) predicts a thermal efficiency of 30.7 per cent. The measured thermal efficiency was 11.4 per cent less, or 27.2 per cent.

2. When the estimated indicated mean effective pressure is 79 psi, the corresponding intake pressure of 7.5 psia predicts a thermal efficiency of 29.7 per cent. The measured thermal efficiency was 9.8 per cent less, or 26.8 per cent.

Although the test data are only approximate, satisfactory correlation with computational values was observed, and this helps substantiate the use of computed prediction information. Since the air-fuel ratios were not similar and instrumentation was rather crude, no effort was made to obtain a direct comparison of volumetric efficiencies; however, additional work should indicate a similarity.

A prediction of anticipated improvements may be useful for comparison purposes if obtained for the usual operating range of a sample automobile. The Cadillac "Eldorado" was clocked at 113 miles per hour at the 1956 Daytona Beach speed trials (31). If the total drag on the

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*Refer to Appendix A, Table 9 for test data taken from 1948 model Chevrolet engine. Computed prediction information is tabulated in Appendix A, Tables 3, 4, 5, 6 and 8, also curves plotted in Figures 2, 3, and 4.

automobile is assumed to be composed of a constant rolling resistance and a wind resistance which is proportional to the square of speed, an approximation of power required to propel the automobile may be obtained (32):

Engine Power =

(Rolling friction constant) x Speed + (Wind Constant) x $(\text{Speed})^3$

If it is assumed that rolling resistance is 25 pounds per 1000 pounds of automobile (the Cadillac weighs 4430 pounds) (33) and if the engine was developing maximum rated power at maximum speed, the constants of the power equation may be solved to obtain the relation:

Horsepower = 0.296 (Speed, m.p.h.) + 0.000194 $(\text{Speed, m.p.h.})^3$

The relatively low power requirements are noted at moderated speeds from the following tabulation:

Car Speed	Required Power
40 m.p.h.	24.2 horsepower
50 m.p.h.	39.1 horsepower
60 m.p.h.	59.8 horsepower
71.5 m.p.h.	92.0 horsepower
113 m.p.h.	305. horsepower

The speed-power relation seems reasonable if it is noted that a 92 horsepower Chevrolet engine should drive the Cadillac at a feasible 71.5 miles per hour (34).

In order to draw conclusions from the data of the Cadillac engine over the operating speed and power range, a limiting parameter must be established. The reference engine used 96 octane fuel with a compression ratio of 9.75:1 (35). Laboratory tests indicate that the highest useful compression ratio for this fuel is approximately

7.25:1 (36, 37). The unsupercharged aircraft engine specifications, Table 7(b), tend to verify this; however, the indicated limit seems confined to relatively low speed engines (38, 39, 40). Increased engine speeds, high turbulence and compact chamber design, and improved cooling of usual "hot spots" in the chamber tend to control detonation in current automobile engines (41). Present designs carry compression ratios as high as 10:1 with engine speeds averaging around 3000 rpm for maximum torque. Since no definite parameters are available, it will be assumed for this study that the Cadillac "Eldorado" engine is operating at the maximum allowable temperatures and pressures for all engine speeds up to its peak torque. This engine was chosen since an examination of Table 7(a) indicates that it reaches the maximum brake mean effective pressure of all current model stock automobiles listed (165 psi at 3200 rpm).

From the work of Taylor, Leary, and Diver (42), it is found that maximum temperature and maximum pressure are the limiting factors for detonation at a given set of operating conditions. Although their test data cannot be applied directly to the computed data due to differences between test and actual operating conditions, one may apply similar principles to the evaluation of other examples. A study of the computational data and curves reveals that the greatest deviation in maximum temperature for a given mean effective pressure is 2.6 per cent over the range of conditions investigated here. It is noted that equivalent indicated mean effective pressures occur at proportionately lower manifold pressures as compression ratios are increased; hence the effect of increased maximum temperatures with higher compressions is minimized. The case is similar with maximum pressures (P_3); however, in this instance the variation is from 30 to 50 per cent. Based on these observations, maximum pressure will be considered the governing detonation factor, and temperature differences will be neglected.

Since it is assumed that the Cadillac maximum pressures form the limiting case, a table of comparisons may be drawn from the approximate powers assumed for operating speeds where the engine is geared to provide maximum power at maximum speed, road slippage being neglected.

Table 1. Comparison of Mean Effective Pressures and Thermal Efficiencies at Random Points Over 1956 Cadillac "Eldorado" Operating Range.

Road Speed (m.p.h.)	IMEP Develop. (psi)	Maximum Allowable Cylinder Pressure	IMEP Available R_c:9.75 (psi)	IMEP Available R_c:15.0 (psi)	Approx. Thermal Efficiency Increase R_c:6.25 to R_c:15.0	Approx. Thermal Efficiency Increase R_c:9.75 to R_c:15.0
40	33.7	1030 psia at 1665 rpm	166	104	36.4	15
50	45.0	1070 psia at 2080 rpm	172	109	33.9	13.2
60	56.1	1080 psia at 2500 rpm	175	111	32.6	11.9
71.5	74.2	1125 psia at 2975 rpm	181.5	115	32.2	11.7
113 (max.)	156	1130 psia at 4700 rpm	182	---	30.5	11.6

It can be seen from Figure 4 that the maximum pressure (P_3) for a ratio of 6.25:1 does not reach the maximum allowable magnitude for the specified fuel at any computed manifold pressure, and it is therefore conceivable that such an engine is limited only by its "breathing" ability where the maximum conceivable indicated mean effective pressure is 197.7 psi for ideal valve timing and frictionless gas flow where intake pressure (P_1) would necessarily be atmospheric 14.7 psia. Observe that the actual power requirements and speeds involved are not at all critical in the calculation of Table 1 since the percent increases in thermal efficiencies for increasing compression ratios are fairly consistent over the entire lower power ranges considered in this investigation. It is concluded that an increase in compression to a ratio of 15:1 in the Cadillac engine would be entirely feasible for all constant driving loads in the usual operating range of moderate speeds and accelerations (note the maximum mean effective pressures allowable relative to those pressures actually used at steady car speeds). It is only for high speeds and accelerations that allowable pressures are exceeded with the 15:1 ratio. An interesting feature is found when a volumetric efficiency of 85 per cent is assumed for an engine with 6.25:1 compression ratio. The maximum attainable indicated mean effective pressure from Figure 3 is 172 psi. At 30 miles per hour, this would allow an increase in maximum available power of over 10 per cent. At the optimum compression ratio for maximum mean effective pressures, this value should be substantially increased.

At this point it is appropriate to introduce the concept of a variable compression engine. The "Eldorado" engine which appears to represent the best in current automobile designs has been chosen as a standard, and increases in efficiencies of over 13 per cent have been predicted over its normal operating range where a maximum compression ratio of 15:1 has been arbitrarily selected from practical considerations as will be shown in Chapter III. At no time was the maximum allowable pressure reached within a moderate operating range of less than 70 miles per hour. This indicates that the specified engine is either uneconomically large, or alternatively, that the 96 octane fuel provided is a good deal more than adequate for normal use. If the 96 octane fuel is replaced by a cheaper fuel of low octane rating and a variable compression ration engine, performance may conceivably be improved over the entire range, if better volumetric efficiencies are available to utilize the increased latitude in limiting combustion pressures. More conservative valve timing may be used to improve low speed volumetric efficiencies. Although the efficiency of the Cadillac "Eldorado" engine can only be improved approximately 13 per cent based on the use of premium gasolines such as Amoco or Sinclair which are currently selling for 34.5 cents per gallon in the Atlanta area, cheaper operation may be expected from low-priced fuels. Typical of cheaper fuels being consumed in lower performance engines are Hood's and People's regular gasoline selling for 28.0 cents per gallon and less. The energy contents of the fuels will be essentially the same (43); therefore a substitution of the cheaper products may reduce fuel costs by 19 per cent, assuming carburetion is unimpaired. The overall decrease in operating costs should be estimated by the total of the efficiency improvement (13 per cent) from increased compression ratios and the cost decrease in necessary fuels (19 per cent). The potential saving of over 30 per cent in the normal operating ranges with possible power gains available for higher loads justifies the investigation of a variable compression ratio engine design (44, 45).

Rotary Valving.--The overall desirability of a rotating valve cannot be satisfactorily analyzed by a purely mathematical approach; however, supporting evidence is presented in its behalf. The chief advantages claimed for the rotary valve, outlined by M. C. I. Hunter, are as follows (46):

1. "Rotary valves do not require periodic adjustment, and the opening and closing of the ports is positive at all speeds without maintenance."

2. "Their action is, or can be, noiseless, with no limit to speed. The modern poppet valve is remarkably silent but gained only with some sacrifices of power."

3. "The valves require no attention in the way of regrinding and cannot readily be thrown out of tune by misuse."

4. Use of rotary valves facilitates more compact combustion chambers with centrally located spark plugs; hence detonation tendencies are improved, necessary spark advances reduced, and indicated efficiencies increased (47).

5. Thermal stresses in the cylinder head may be reduced due to more uniform heating (48).

6. "Perfect balance of rotors is possible in most designs...so far as the valve gear is concerned the speed is almost unlimited."

7. Valve spring breakages are absent, and there is no tappet vibration.

8. No exhaust valve heads are exposed to form hot spots in the combustion chamber (49).

9. Superior breathing characteristics may be gained by streamlined ports (50).

10. Turbulence may be increased, especially at low engine speeds due to valve action with changing tangential flow of the incoming charge; hence detonation is minimized (51).

11. The elimination of stagnant areas within the combustion chamber reduces difficulties with carbonization.

12. Very low rates of wear may be possible. (Of course this is contingent on good lubrication.)

13. The number of operating parts may be fewer than with the poppet valve.

14. "The compression ratio can be increased and fuel of low octane value used." (52)

15. The specific fuel consumption for rotary valve engines should be less than that for corresponding poppet valve arrangements (53).

In addition to these claims, the proposed design will completely shield the electrodes of the spark plug except for a short time interval during which actual firing occurs (54).*

There is much evidence in the literature to support these claims of rotary valve superiority, as the bibliographic references indicate; however, it is necessary to also recognize the inherent faults in the principle. The principal disadvantages of the rotary valve are discussed by M. C. I. Hunter (55), and the following guide is established toward the design of an acceptable system:

1. Maintenance of close operating clearances around the valve body under all loads to minimize gas leakage and excessive compression losses (particularly at low speeds, high powers, or both)(56).

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*Hot plug electrodes have been established as a definite factor in engine detonation(54).

2. Assured lubrication of working faces of hot rotors so as to avoid undue oil consumption and seizure over the full range of operating conditions. (Excess oil consumption is both expensive and likely to cause fouled spark plugs.)

3. Proper mating and area reduction of rubbing surfaces in order to avoid excessive friction losses (57).

4. A careful evaluation of available fabrication techniques and material requirements in order to control production costs (58).

These weaknesses are also well defined in the literature which deals with the developmental stages of the sliding surface valve.

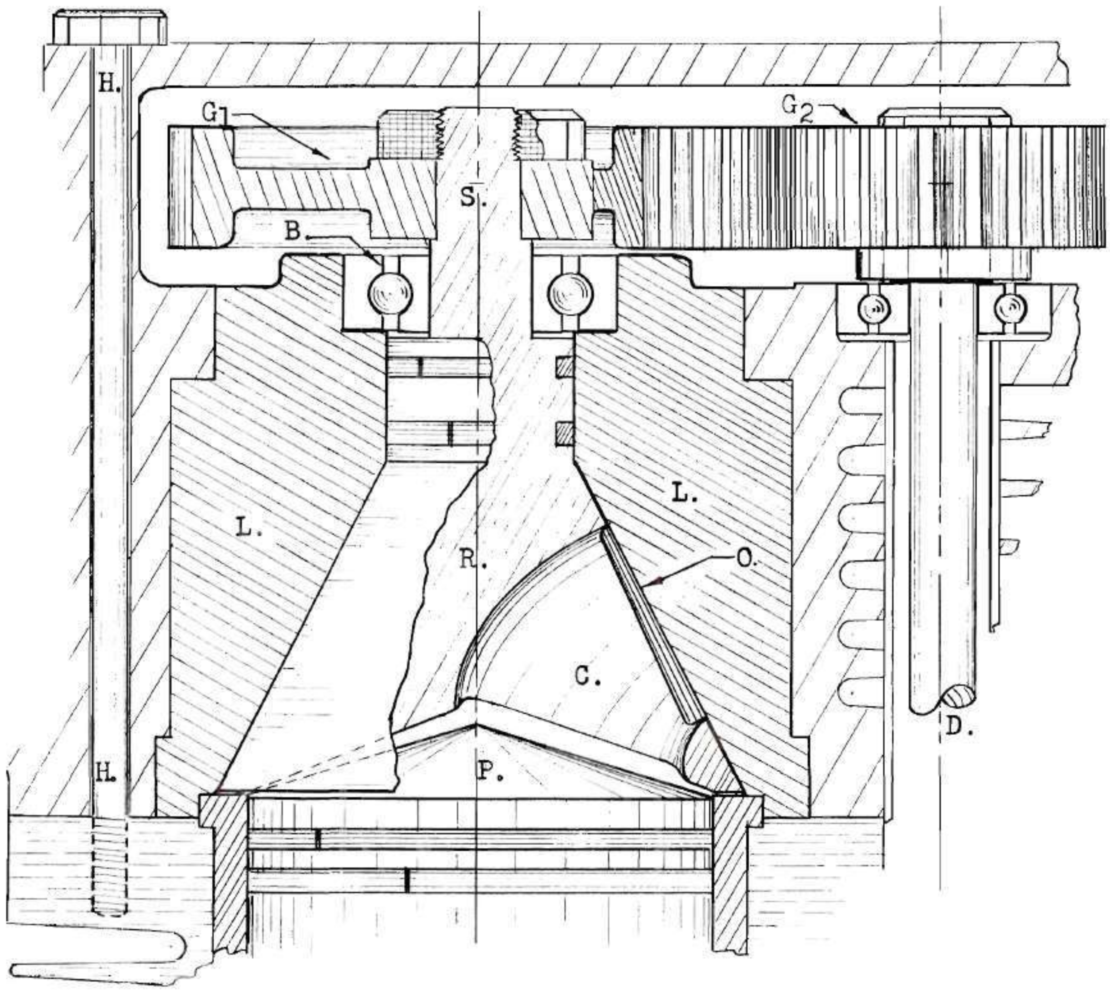
Many engines using the sliding valve principle have been built and tested. A few of the more notable successful models are mentioned and offered here in evidence of the potentialities existing in the rotary valve design.

R. C. Cross has built and tested a number of horizontally rotating valve engines (59, 60). His work, commencing in 1922, has been so significant that the general type of engine with which he experimented has retained his name. Most of the work of Cross involved small single cylinder motorcycle engines of approximately 15 to 21 cubic inches piston displacement. The engines which he finally built were reportedly smooth running, reluctant to detonate, and capable of brake mean effective pressures as high as 154 psi at 5700 rpm. One such engine developed 1.16 horsepower per cubic inch at 6100 rpm using gasoline of 66 octane, yet low speed torque was good (61).

Although there is evidence that his performance claims are excessive, a recent experimenter, Merrit Zimmerman, has made praiseworthy progress in utilizing the experience of Cross (62). Zimmerman has built a two-cylinder motorcycle engine of approximately 15 cubic inches displacement employing modern high strength materials. His tests indicate good reliability and smooth operation using 85 to 90 octane fuels in the 12:1 compression engine. The 230 psi brake mean effective pressure which he claims for engine speeds above 6000 rpm is not feasible; however, since it can be predicted from Figure 3 that this value would call for a supercharged engine. Zimmerman claims the engine will operate at a full throttle 12,000 rpm producing 3 horsepower per cubic inch displacement for approximately four minutes before excessive temperatures cause damage.

A second fundamental type of valve is associated with the name of its prime developer, F. M. Aspin (63, 64, 65, 66). This engine employs a valve that rotates at one-half engine speed about a vertical axis which is usually the generating axis of its respective cylinder. A diagram of this type of engine is presented in Figure 6, accompanied by a tabulation of principal parts. The engine made its appearance in British journals during 1937. It has operated reliably at a 14:1 compression ratio during full power runs at 14,000 rpm for many hours with no apparent difficulties. The performance shows a maximum brake mean effective pressure of 180 psi. The oil consumption was reported comparable to that of other engines at full load. This type of engine was produced commercially in Britain in the form of a small tractor engine. The fuel consumption at full load is an exceptionally good 0.43 pounds per horsepower hour.

The concluding example is presented in defense of sliding type



Essential Parts of Assembly:

B. Radial thrust bearing for drive gear loads.
C. Rotating "cell" type combustion chamber.
D. Drive shaft geared to engine crank shaft.
G_1, G_2. Spur drive gears for valve control.
H. Head and cylinder retaining studs.
L. Fixed valve liner provided with bearing surface for valve contact.
O. Port orifice. (Alternately inlet, spark, and exhaust.)
P. Piston.
R. Rotor body of valve.
S. Rotor shaft.

Figure 6. Typical Aspin-Type Rotating Cylinder Head Valve.

valves. The Bristol "Hercules" and "Centaurus" and the Napier "Sabre" aircraft engines are presently in operational use of the Royal Air Force. They are undoubtedly the quietest aircraft engines yet witnessed by this observer. These engines employ reciprocating and oscillating valve sleeves and attain excellent performance, notably in respect to oil consumption.*

*In Appendix A, Table 7(b), the Bristol "Hercules" and "Centaurus" engines are specified to consume 0.008 pounds of lubricant per horse-power-hour. This is less oil consumption than that of any of the seventeen other engines listed.

CHAPTER III

VALVE DESIGN

Proposed Design.--As indicated in Figure 7, the proposed design will incorporate an extension of the usual cylinder above the top end of the piston travel. Inside this extension a finely machined metal plug is fitted and geared to rotate at one-half engine speed. A combustion "cell" is machined into the rotor, and its lower surface is contoured to match the convex face of a dome type piston.* The rotor may be moved axially for a short distance within the cylinder under action of hydraulic pressure. A constant supply of oil pressure at O_e, Figure 7, acts against the lower surface of the gear (G_2) shoulder providing a steady lifting force on the rotor. From the same source, (O_e), oil is emitted to the upper surface of the rotor body through the reed valve (R). The oil (O_2) carries the entire thrust load from the compression and firing strokes of the engine. The exact vertical position of the rotor is regulated by metering the discharge rate of the oil (O_2) which is expelled through the duct via the metering annulus into the gear box at O_3. It should be observed that while oil (O_1) flows into the system at a steady rate, its discharge (O_3) may be controlled in the metering annulus. If the controller is moved upward (Figure 8), the restriction

*Observe the large quench area, Q, indicated in Figure 7.

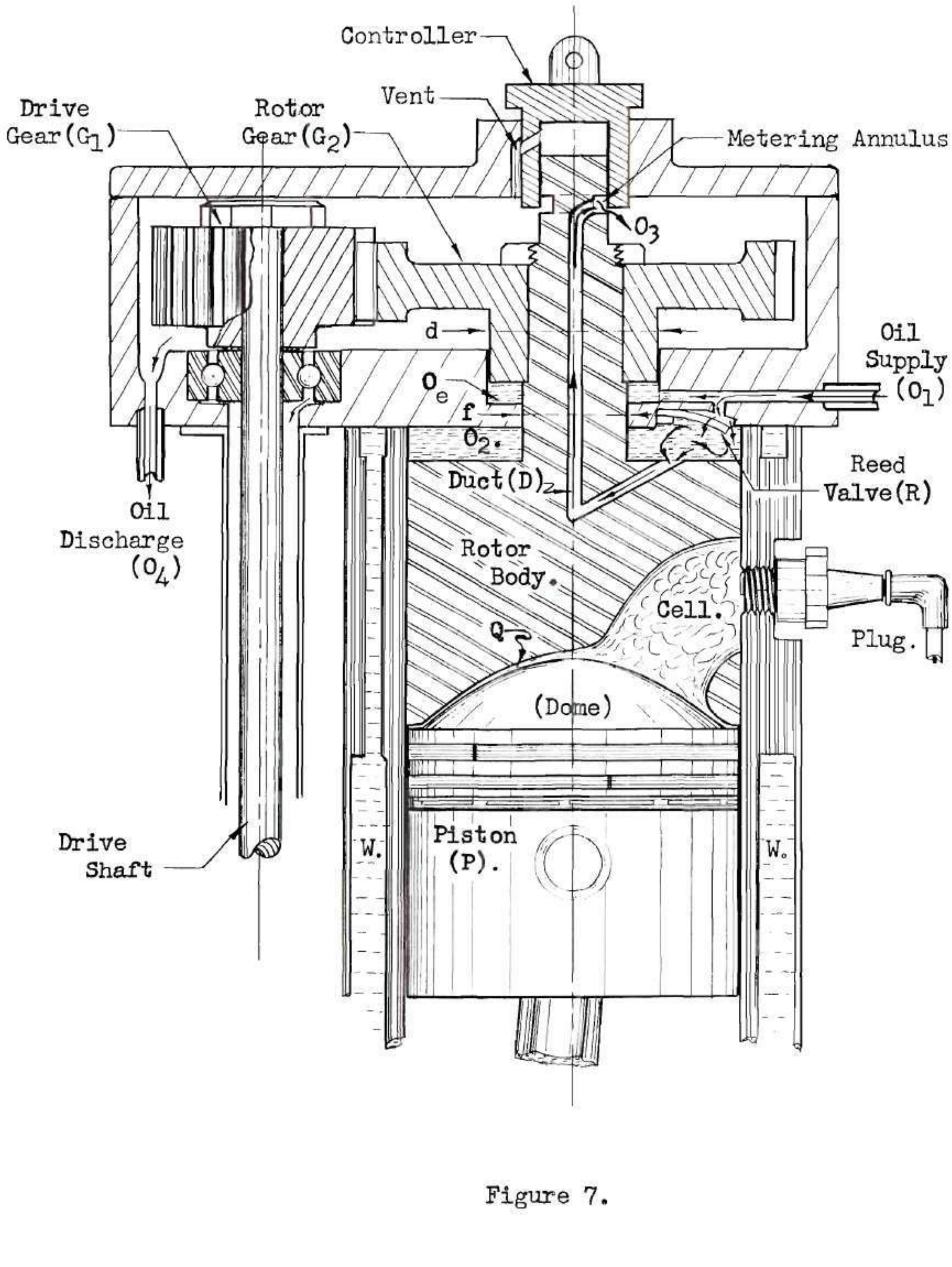


Figure 7.

Proposed Valve and Mechanism

at the metering annulus is reduced and oil from the thrust bearing (O_2, Figure 7) is expelled through the duct under the force of pressure (O_e) lifting the rotor. As the rotor is displaced upward the total volume of the combustion space is increased and the compression ratio therefore lowered.

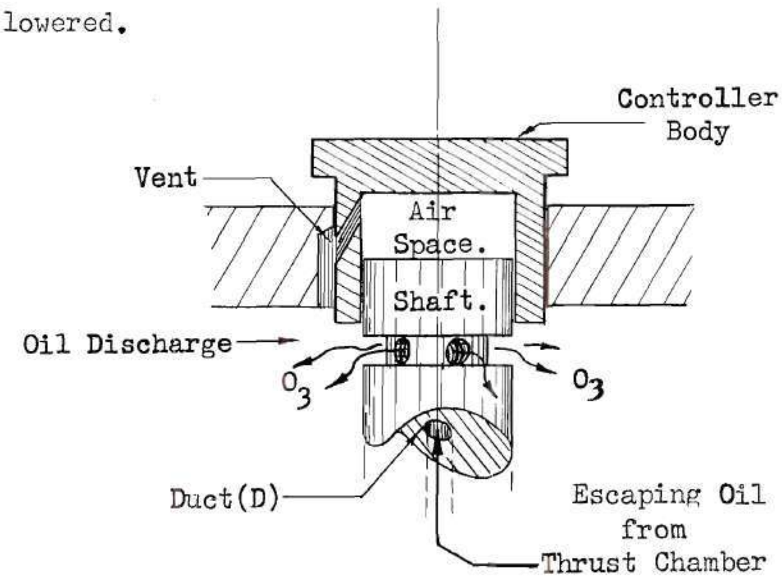


Figure 8. Controller Mechanism in Relief Position.
(Raising rotor reduces R_c)

Similarly, if the controller is moved downward, the metering annulus will be constricted and oil flowing into the thrust chamber (O_2, Figure 7) will force the rotor downward, thus raising the compression ratio of the engine. The oil (O_1) flowing into the system is provided by positive displacement gear pump and hence must circulate constantly. Note that the reed valve (R , Figure 7) acts to provide a pressure differential between the large and small thrust chambers. This valve facilitates the use of a single oil supply source since the oil pressure in the small chamber (O_e) must be sufficient at all times to raise the rotor when a reduction in compression is desired and to support the rotor against the force of gravity between compression and firing intervals. A third desirable feature of the smaller thrust chamber is the positive pressure which it affords to the oil at all times. This pressure head guards against possible vapor locks within the system, particularly when the unit is idle.

Engine Lower Unit and Auxiliaries.--For purposes of simplicity in data, analysis and machine operations, it is desirable to employ a single cylinder engine for developmental purposes. A British built Velocette motorcycle engine, model MSS, motor number 6351, was chosen in this particular instance from personal experience of its durability. This engine is especially desirable since it has an extremely rugged lower end (67). Noteworthy are the short rigid crank, forged steel connecting rod and roller bearing suspension and crank throw (Figure 9). The original dimensions of the Velocette were 81 mm (3 3/16 in.) bore and 96 mm (3.78 in.) stroke for a displacement of 495cc (cubic centimeters) or 30.25 cubic inches. In order to partially compensate for increased compressions, a smaller 3 1/16 in. aluminum alloy piston was selected from a standard Ford automobile for initial tests at lower compression ratios and engine speeds under 4000 rpm. The displacement will then be 27.8 cubic inches. The Velocette engine is equipped with a Bosch type K.C.1 - N4 magneto for positive ignition. The parts required to adapt the Velocette to a test engine include:

1. Cylinder piston assembly
2. Valving apparatus

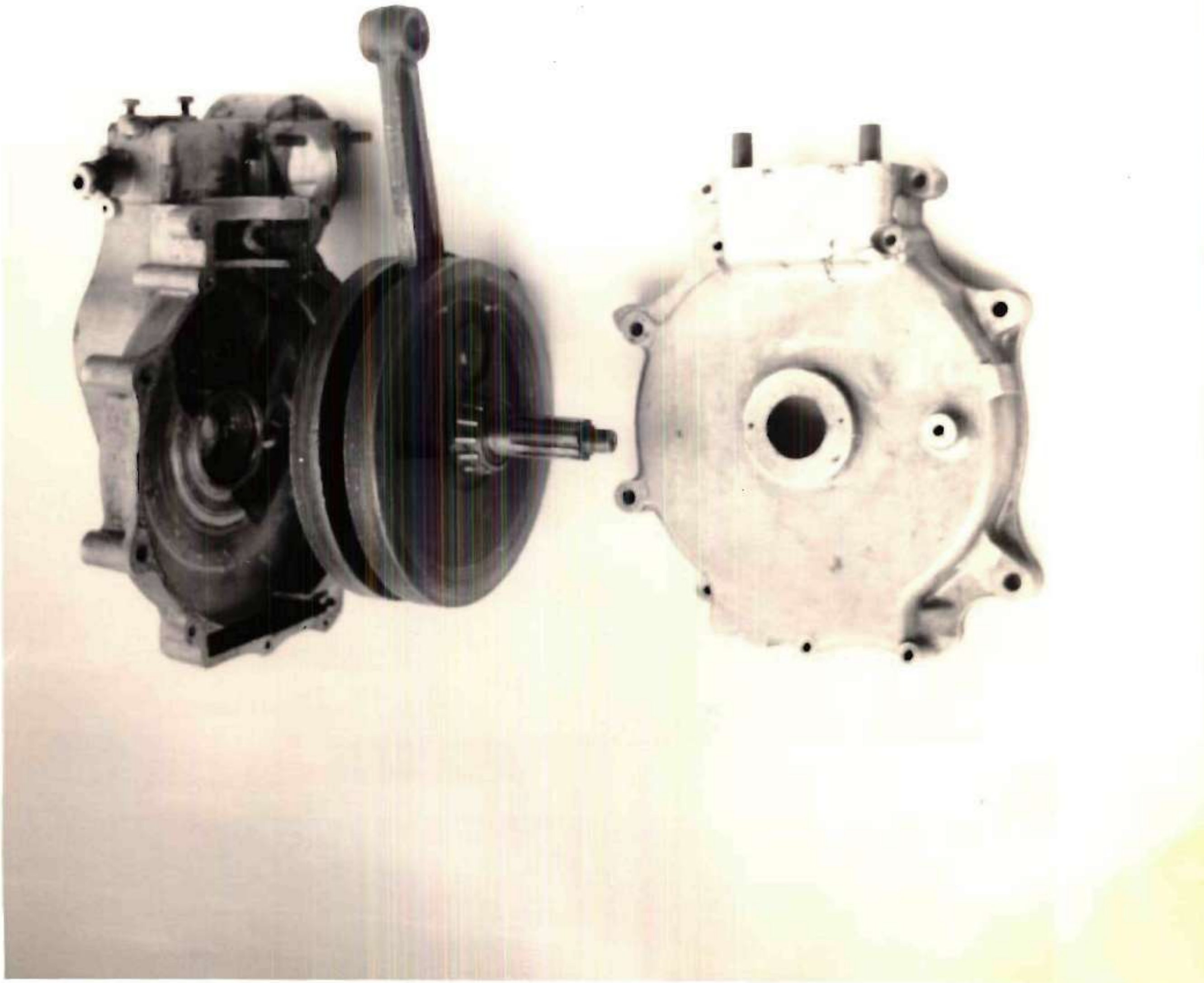


Figure 9.

Velocette MSS Crankcase and Crankshaft Assembly

3. Variable compression control
4. Valve drive apparatus

Valve Timing.--Due to geometric considerations, it is necessary to allow equal opening periods for both intake and exhaust ports if maximum capacity is to be derived from the valve. A typical specification is selected from review of the rotary valve literature (68): The inlet port opens 12° early and closes 48° late; the exhaust port opens 48° early and closes 12° late. These figures are quite conservative when compared to present automobile specifications (Appendix A, Table 7c). For example, the current Cadillac engines hold the inlet port open for 324° rotation of the crank shaft whereas the proposed engine will only employ inlet opening during 240° rotation.

Valve Port Dimensions.--In order to realize minimum gas friction through the ports, the inlet and exhaust ports in the cylinder walls should be machined to match the side opening of the combustion cell in the rotor. In order to obtain the timing specified, the centers of the exhaust and inlet ports must be oriented respectively 126° clockwise and counter-clockwise from the position of the rotor port when the engine piston is at the top dead center. The chordal width of the ports is computed to be 1.048 inches.

If the ports are rectangular in shape and a length of 1 1/2 inches is used, the area of the valve opening is 1.57 square inches. This valve area is related to the piston area by a 0.213 ratio. The intake (larger) valve head to piston area in the Cadillac being considered is similarly related by a 0.191 ratio. Since the throat of the of the poppet valve port must necessarily be smaller than the corresponding valve head and since the valve stem restricts part of the effective passageway, the valve area of the Cadillac cannot be as effective as the proposed rotary valve port area. Some sources indicate that sliding valve port areas may possess as much as twice the efficiency of equivalent poppet-valve areas (69). At any rate, the port being designed should be satisfactory for test purposes and practical considerations of port sealing and combustion cell dimensions discourage the attempt for more area at this stage of design. (Less area might well be considered.)

Valve Sealing.--Two primary problems are recognized in connection with sealing. Gas pressures within the cylinder must be controlled to prevent excessive leakage around the inlet and exhaust ports (leakage into the oil chamber presents no problem since the oil is always at a higher pressure than the gas), and the leakage of lubricating oil into the combustion space must be regulated. It is proposed that a spring loaded brush type pressure device be used for sealing purposes around the ports. The port would be machined into the end face of a cylindrically shaped section of low friction bearing stock, and this face in turn cupped to fit snugly against the rotor. Copper alloys containing lead and tin appear to operate satisfactorily against most ferrous rotor materials (70), especially under poor lubrication conditions. Also, their high thermal conductivities are valuable in facilitating heat dissipation. Phosphor-bronze, SAE 64, is recommended for the first sealer to be experimented with. The proposed seals are illustrated in Figure 10.

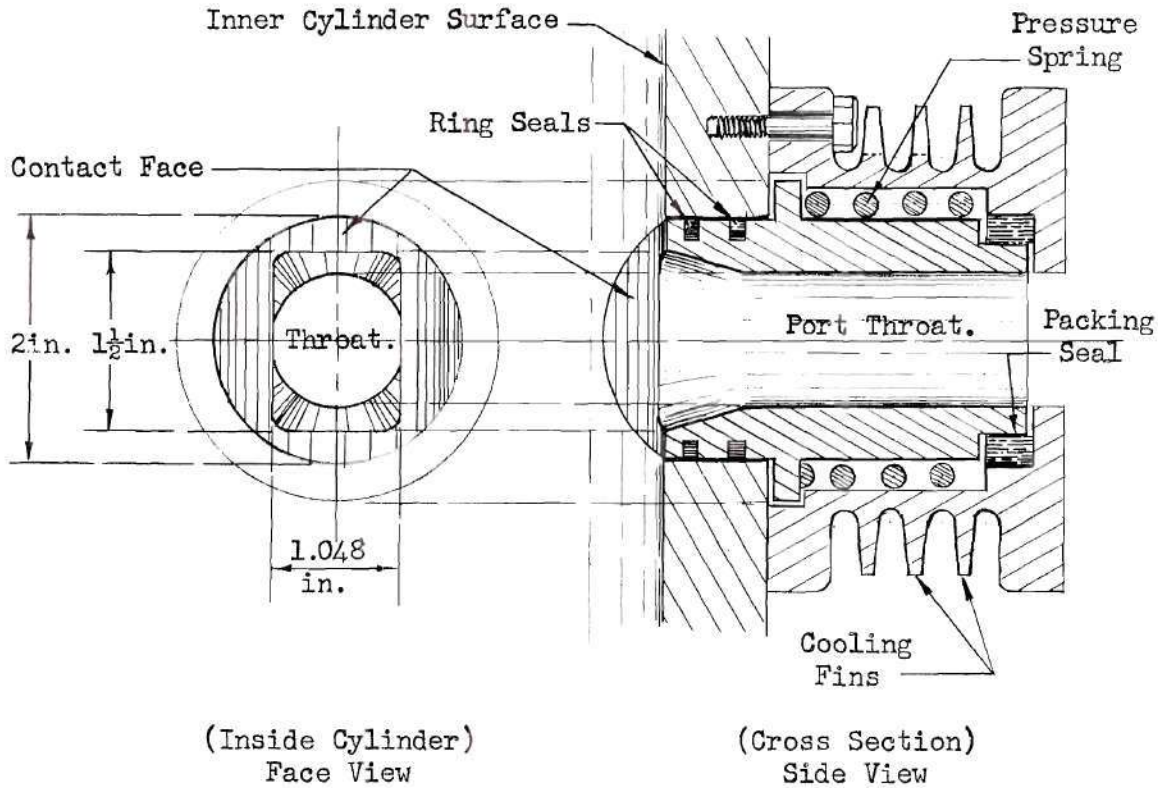


Figure 10. Brush Type Spring Loaded Port Sealer.

The outer diameters of the brush sealer bodies should be machined to two inches and carefully fitted to their respective passages through the cylinder block wall. It is suggested that the sealer bodies be machined to approximately 0.002 inches oversize, then finely threaded on a lathe to a depth of 0.005 inches. A standard 64 threads per inch should work satisfactorily. This feature allows for a forced fit in the cylinder and self adjustment to thermal expansion. Two 2-inch diameter, 3/32 x 3/32 inch expansion rings are fitted into each of the two cylinder block holes and then stationed in machined grooves of the sealer body (Hastings gray iron piston rings, number 275, appear to be satisfactory for this task). These rings, along with close machine fits of the sealers in the cylinder, are provided for gas leakage control around the pressure elements. A study of the calculations in Appendix B reveals that exhaust pressure acting against the partly exposed face of the sealer body might compress the pressure spring during operation, causing the sealer body to chatter against the rotor.

Pressure springs may be designed accordingly through application of the formula (71):

$$\text{Spring Constant (lb./in.)} = \frac{(\text{Diam. wire, in.})^4 \times (\text{Shear modulus, psi.})}{64(\text{Radius of coils, in.})^3 \times (\text{No. effective coils})}$$

It is determined that a helical coil of No. 3 (Washburn and Moen gage) (72) spring steel may be employed to provide sealer pressures. Inconel X, A.S. 140, is desirable spring material since it can satisfactorily withstand high operating temperatures (73). The helices should be wound to fit snugly but not bind on the 2-inch diameter sealer bodies. Ends should be squared and ground (74). A standard spring of 4.75 total turns and 1.5 inch overall length may be compressed 0.137 inches to provide a recommended 25 pounds pressure on the exhaust sealer and a 0.055 inches for a recommended 10 pounds pressure on the intake. The relatively high spring constant of 183 pounds per inch is considered desirable because of the high speed vibrational characteristics (75) and because of the ease of obtaining adjustment through shims. The sealer housing may be cast from aluminum and machined as necessary. The housing should be finned to help cool the springs.

Thrust chamber oil control will employ the principles of pressure drop across labyrinth type seals which circle the rotor and cylinder as indicated in Figure 11 (76). A limited quantity of oil must necessarily leak through the bearing clearance around the upper shoulder of the rotor.

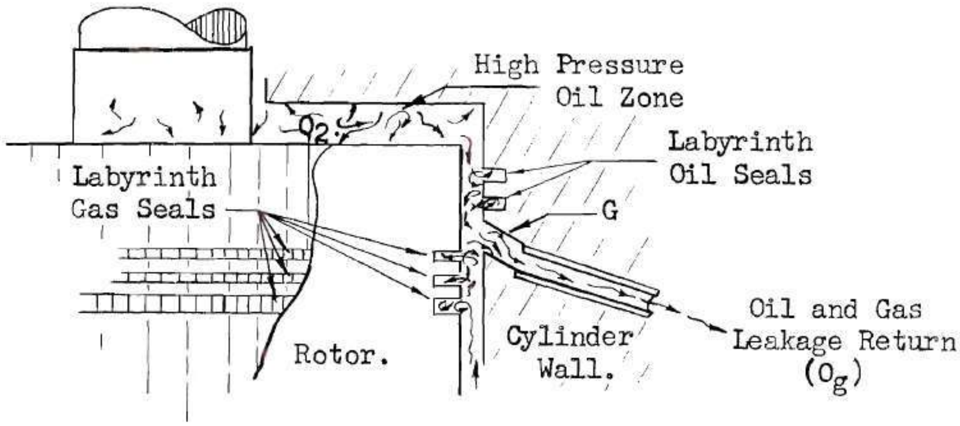


Figure 11. Compression Chamber Oil Control System.

This oil is finally trapped in chamber G, Figure 11, and under the influence of a reduced pressure from slight cylinder gas leakage, induced to return through a passage in the cylinder block to the oil reservoir (and cooler).

Lubrication.--Since the upper surface of the rotor and adjacent cylinder faces operate with hydrodynamic lubrication and since all parts of the drive mechanism operate in oil discharged from the thrust chamber, the only major problem apparent is the lubrication of the port sealers and lower rotor surfaces. This problem is very critical, since excess lubrication may foul the spark plug and cause undue formation of sludge and carbon deposits in the combustion chamber with accompanying losses in economy. Conversely, insufficient lubrication may cause serious damage to the engine. It is hypothesized that the proposed oil control system is adequate to avoid excess lubrication. The principal hazard is therefore reasoned to be insufficient lubrication of the lower rotor surfaces. This characteristic can only be debated until actual testing of the engine, but it is believed at this point that sufficient lubrication will leak from the oil return chamber (Region G of Figure 11) down around the pressure sealers where it will be distributed sufficiently to lubricate all necessary surfaces. No attempt will be made to recover this oil in the initial test engine; however, it may later become necessary to add a scraper system if the need becomes obvious.

Oil Control for Compression Regulation.--The function of the hydraulic control system has been explained earlier. In order to assure support of the rotor against gravity and intake vacuum and to aid in expelling oil from the thrust chamber when the need arises, adequate oil pressure (O_e, Figure 7) must be assured at all times through proper design of the reed valve (R, Figure 7) and the lower thrust shoulder of the rotary valve gear. If it is assumed that 5 psia minimum pressure may be obtained in the cylinder while the engine is operating at a barometric pressure of 14.7 psia and (as will be shown later) the weight of the rotor assembly is 8.04 pounds, the maximum force downward to be supported is computed to be 78.4 pounds (neglecting the weight of oil in the lower thrust chamber). For reasons of strength and gear support, the rotor shaft diameter (f, Figure 7) was taken as 0.92 inches. As a conservative allowance, the outer gear hub diameter (d, Figure 7) is selected as 1.75 inches and the oil pressure, O_e, an average of 80 psig. This will provide a net lifting force of 157 pounds. Since the exposed thrust surface on the top of the rotor is 6.70 square inches, an oil pressure (O_e) of 23.5 psig must be present to hold the rotor in equilibrium. This pressure is considered adequate to retard oil vapor formation and also provide pressure for rapid response from the discharge controller valve (Figure 8). The oil is to be supplied by a conventional gear type pump equipped with a ball check relief valve limiting pressure (O_1) to 80 psig. The reed valve (R, Figure 7) is designed to provide a pressure differential of 50 psia.* This will allow the entire control system to operate from the same lubricant supply source. A total pressure throttling of 6.5 psi will be allowed through the rotor duct and metering annulus. Obviously the pressure (O_2) in the lower thrust chamber will far exceed the 80 psig supply pressure during the firing stroke of the piston (O_2 equals 548 psia when the cylinder pressure reaches 500 psia). However, the reed valve provided operates as a high speed check which protects the supply system from excess back pressure. Rapid response in reducing compression is essential if the engine is to be able to assume sudden heavy loads at moderate speeds. The computed 6.5 psi pressure throttling is not considered adequate control margin in itself; however,

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*Appendix C presents the details of the reed check-valve design.

cylinder thrust pressures will greatly amplify this value and aid in providing accelerated response. The discharge duct should be as large as practical to reduce flow resistance.*

Materials Selection.--In line with the experience of earlier sliding-type valves (77, 78), the material for the rotor was selected to be Nitralloy 135, aircraft specification, oil quenched from 1725°F and tempered at 1200°F. The surfaces of the rotor should be nitrided for approximately 72 hours at 975°F after all machine work except balancing has been completed (79). It is desirable to protect the upper surface of the rotor shoulder from hardening by coating with "Sel-Nite" paint or a thin tin electroplate (80). This will leave a surface where balance corrections may be made by normal machining operations. These specifications will provide a tough rotor core capable of an 84,000 psi tensile endurance limit combined with approximately 0.018 inch of wear resistant surface with a hardness number of 95 on the 15-N Rockwell scale. The nitriding process is especially desirable since operating temperatures of 1000°F are permissible without appreciable softening of the surface. A diameter growth of 0.002 inches is predicted from the specified surface treatment. A cylinder material of mild carbon steel should provide a satisfactory mating material with the rotor and will facilitate the machining operations (81). Standard 3 inch inside diameter by 5 inch outside diameter, annealed, SAE 1018 tubing was

*The oil discharge duct through the rotor shaft should first be tried at 3/16 inch diameter; however, operating tests may indicate a larger size is desirable.

selected in this instance. Any commercial cast aluminum, such as Alcoa 195 should provide adequate strength for the cylinder head and gear housing (82).

<u>Thermal Expansion and Working Clearances</u>.--The coefficients of thermal expansion of cylinder and rotor are essentially equal at 0.0000128 inch per degree centigrade (83). Predicted mean temperature of 500°F for the rotor skirt and 200°F for the cylinder are based on experimental information in the literature concerning analogous arrangements (84, 85, 86, 87, 88). An arbitrary maximum of 700°F will be assumed to cover unpredicted difficulties which may arise during operation. The difference in expansion of the rotor and the cylinder is therefore computed to be 0.00356 inches. A clearance of 0.005 inches is recommended for the rotor skirt to allow for expansion, oil clearance and machining tolerances. Since the rotor receives a large part of its cooling effect from oil circulating in the thrust chamber, consideration of the thermal gradient which must exist makes reasonable the assumption that the rotor top is no more than 100°F warmer than the cylinder walls. This indicates that only 0.0007 inches need be allowed for thermal expansion. In view of oil control considerations in the thrust chamber, the top 0.75 inches of rotor should be allowed 0.002 inch clearance from the cylinder bore. The entire cylinder contact surface of the rotor should be lapped to final dimensions using conventional techniques (89). The surface roughness of approximately 2 microinches so obtained may be expected to reduce galling tendencies noticeably.

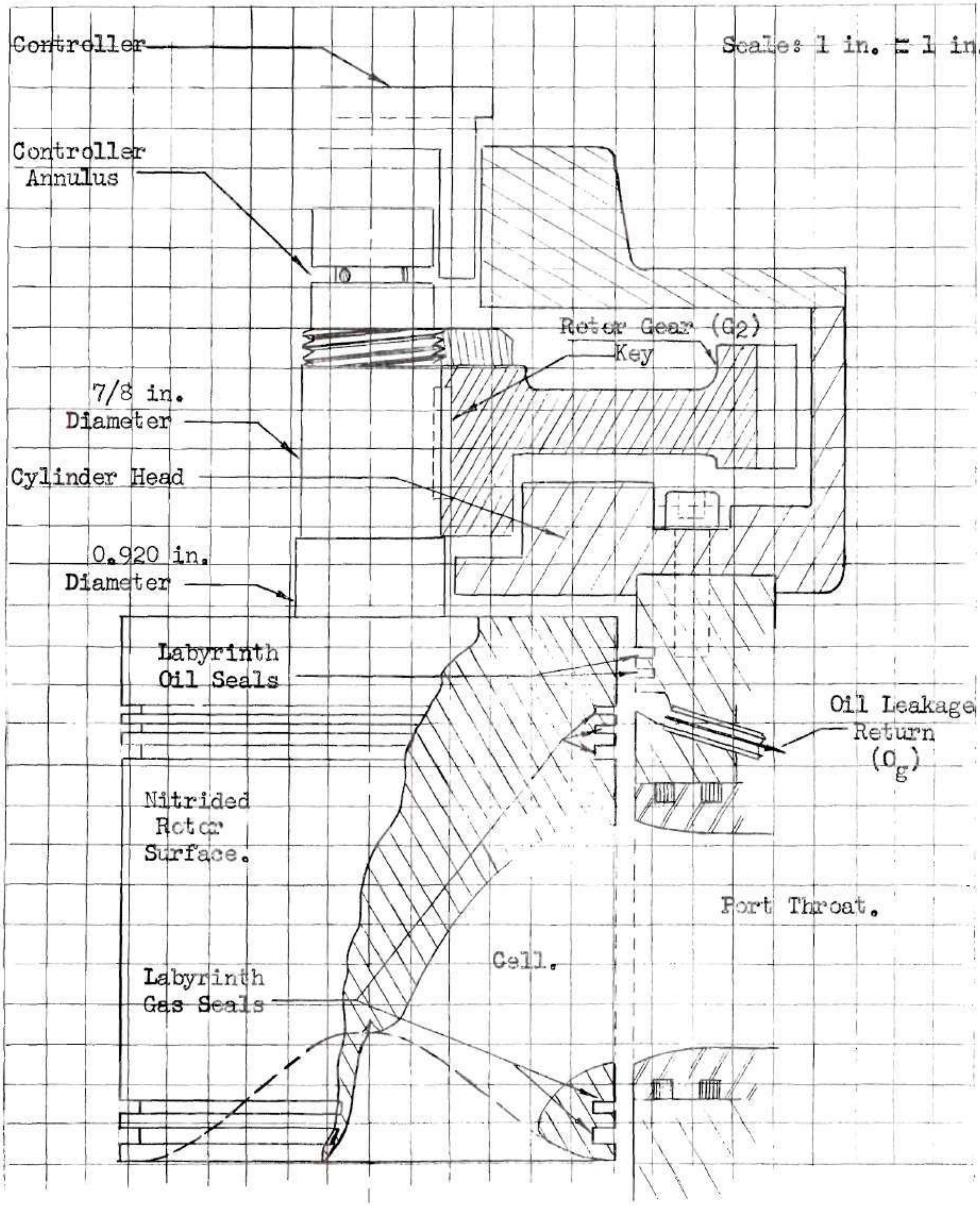

Figure 12. Detail View of Rotor, Upper Cylinder, Cylinder Head
and
Control Mechanism

Selection of Dimensions.—The dimensions and proportions of the valve are based on considerations of necessary adjustment of rotor position, specified port dimensions, reasonable allowances for oil and gas seals, optimum chamber dimensions for desired compression range and good breathing, compression turbulence, heat dissipation and oil friction. The details of the proposed mechanism are shown in Figure 12. Dimensions of the system are drawn to full scale with working clearance exaggerated for clarity.

A rotor was machined to the dimensions indicated in Figure 12 and the design will proceed from this point in the form of an investigation into the kinetics of balancing this rotor and adjusting the torsional vibration which may be excited in its drive mechanism.

Static and Dynamic Rotor Balance.—All vibrational advantages claimed for this rotary valve engine are dependent upon an accurate balance of static and dynamic radial forces within the rotating mass. The relative importance of proper balance appears obvious when the 7.1 pound centripetal force necessary to constrain a mass of one ounce (1/16 pound) to motion in a circular path of one inch radius at the speed of 2000 revolutions per minute is computed (90). At 4000 revolutions per minute, the force will be increased by a factor of four. Since one ounce of the rotor material occupies only 0.223 cubic inches (91), it becomes necessary to account accurately for the eccentricity of mass distribution incurred in the machining of the combustion cell and port. Since the contour of the port was hand sculptured from the solid rotor stock, proportions emphasize smooth uninterrupted contours and gradual changes in cross sectional areas.

The nature of the calculations involved in combining an aerodynamic port shape with a cavity of ideal combustion dimensions is so complex that the problem can only be reduced by physical research. Any attempt to specify exact mathematical generatrices at this stage of development is considered to be imprudent. Since the external rotor dimensions have been established, the problem of accurate balance may be met on any conventional dynamic balancing machine. However, the drill holes characteristic of experimental balancing techniques are undesirable from the standpoint of heat blocks and oil turbulence.* In this light, the rotating mass will be investigated in order to effect an approximate balance which features the refilling of all weight compensation recesses with materials of appropriately higher or lower density. It is convenient to initiate this investigation by locating the center and quantity of mass removed eccentrically from the rotor in machining the port. This was accomplished experimentally as described in Appendix D. The total mass removed in forming the port amounted to 0.435 pounds, and it was located at a radius of 0.971 inches from the rotor axis and a longitudinal distance of 1.064 inches from its skirt. Armed with this information, an analytical balance was computed through a process of trial and error evaluation of various geometric arrangements of material substitution in the rotor. There are numerous methods of establishing such a balance; however, the solution presented

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*It is considered undesirable to mar the contours of the combustion chamber or to interfere with the accurately machined cylinder bearing surfaces of the rotor. All balance compensations made to the rotor itself must therefore be effected through its upper shoulder about the shaft as indicated in Appendix D, Figure 18.

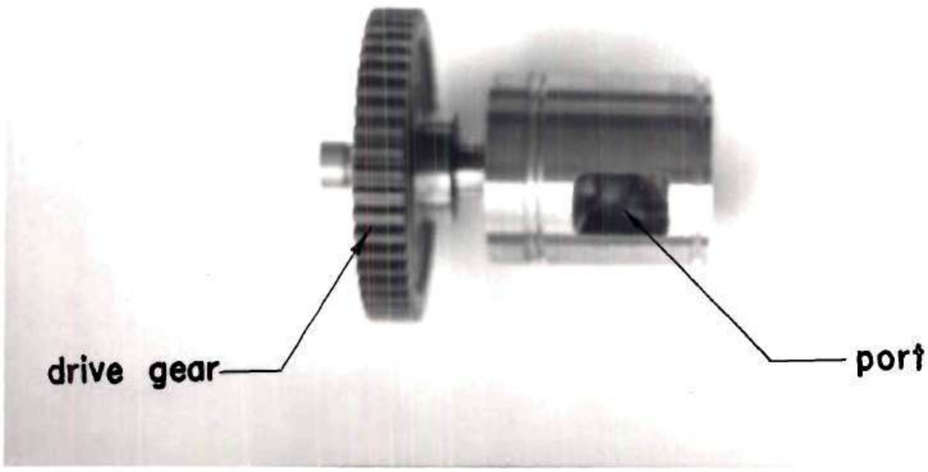
in Appendix D is recommended on the basis of simplicity attained through utilization of already existent features of the rotor and drive assembly. This arrangement employing three aluminum plugs, two lead filled copper plugs and appropriate weight removal from the drive gear is especially advantageous from a standpoint of heat dissipation through the large bulk of steel in the rotor body. The bulk of the plugging material was aluminum and copper with respective thermal conductivities of approximately five and eight times that of the steel base metal (92). It was advantageous to gain as much weight as possible near the point where material was removed in the port, hence lead was a desirable plugging material because of its high density. In view of its low melting temperature of 621.2°F and relatively poor thermal conductivity, the concentric arrangement of lead within the protective copper plugs appears advisable. A physical bond was obtained by holding the copper container at elevated temperature while molten lead was poured inside. Any common soldering flux may be helpful in this process.

It is difficult to approximate the error to be expected in computations of this type. The advantage of extreme mathematical accuracies is questionable in view of the experimental means employed in establishing the initial conditions of unbalance. At any rate, the results cannot be expected to be exact, and small errors should be compensated for on any sensitive, conventional type of dynamic balancing machine. The compensation holes should be relatively small at this point, and they may be drilled at any point on the upper shoulder of the rotor or in the drive gear web. The density of oil to be used in the thrust chamber must be considered when recesses are drilled in the rotor.

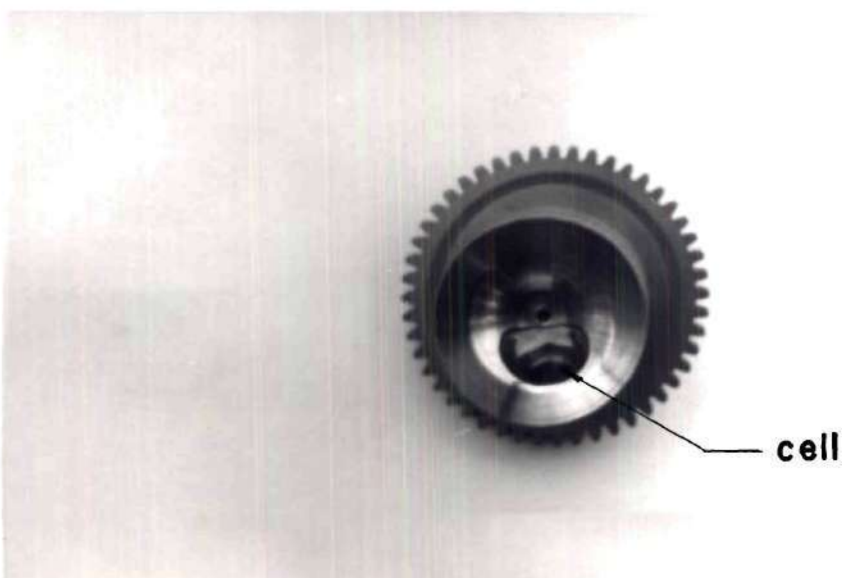
Vibration Analysis and Design of the Rotor Drive.—The ever-present forcing functions provided by intermittent power impulses and inertia effects from reciprocating components necessitate concentrated attention to engine vibrations. The complete elimination of all vibrations is a practical impossibility; therefore, we must adjust the design so as to minimize harmful effects. The Velocette engine was originally balanced during manufacture, and the only consideration which will be given to components of the "lower end" in this analysis is to the preservation of their original weight specifications. The new piston assembly weighs 0.955 pounds and should carry a ballast weight of 0.085 pounds to restore factory balance. Since a new bronze bushing must be machined for the upper end of the rod to accomodate the smaller 0.750 inch diameter piston pin, adequate compensation may be obtained by adding a small shoulder to the bushing.

The rotor and drive gear pictured in Figure 13 were suspended in a bifilar pendulum apparatus and caused to oscillate about their normal axis of rotation. The system oscillated at a natural frequency and through careful measurement of the corresponding period, the inertia of the composite mass was determined (93).* The inertia of the rotor and gear was corrected for pendulum holder mass and amounted to 0.03584 inertia units. By a similar type experiment involving the internal support of gears on a

*See Appendix E for description of experimental procedures employed in the determination of inertia.



Side View



Bottom View

Figure 13.

Rotor and Drive Gear

knife edge and appropriate analysis of their pendulum action, the inertias of the remaining gears were measured (94).*

Torsional vibrations are detrimental to the rotor drive linkage and are likely to create oscillating stresses far in excess of the frictional drag to be overcome in the rotor (95). Considering the availability of standard products and existing space limitations, the highest capacity gears convenient were selected for the drive mechanism.** Further refinement in gear specifications will be postponed until actual operation of the mechanism can be evaluated. Since the rotor must be powered by spur gears in order to allow for adjustment of compression, it it most convenient to drive the valve mechanism by a vertical shaft coupled to the crankshaft on the lower end by bevel gears. The torsional flexibility of this rotor drive shaft must be adjusted in order to keep the natural frequency of vibration between the rotor mechanism and engine crank sufficiently high to avoid excitation by any of the lower orders of power impulses from the engine (96). The inertias of both spur gears (G_1 and G_2, Figure 7) and the rotor may be lumped into one equivalent rotating mass by taking the 2:1 gear ratio into consideration. Since the crankshaft speed is a convenient reference, the inertia of the rotor and its drive gear was corrected with respect to the crankshaft. The summation of the inertia of the smaller spur gear and the equivalent inertia of

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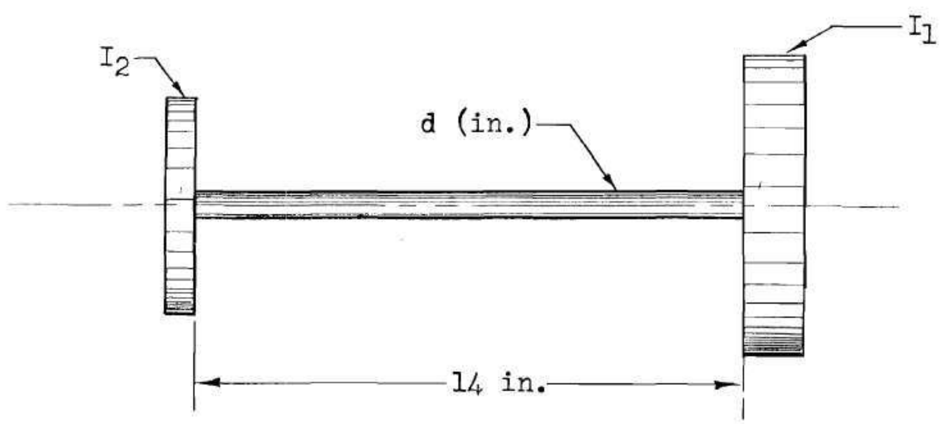
*See Appendix E for description of experimental procedures employed in the determination of inertias.

**See Appendix E for specific description of drive components and vibrational analysis.

the rotor and its gear may be taken as the equivalent moment of inertia of a single mass attached directly to the upper end of the rotor drive shaft. The inertia of this equivalent mass is computed to be 0.012118 pound-inch-seconds squared. Note that the flexibility in the short length (1/2 inch) of shaft connecting the rotor and its drive gear is neglected in view of the more critical 14 inch rotor drive shaft which will be investigated.

The crank end of the engine was broken down into approximately seventeen fragments which were analyzed individually for rotational inertia. The inertia summation of these individual elements provides an approximate equivalent moment of inertia for the entire lumped mass of 0.52595 pound-inch-seconds squared. The vibrational characteristics of the rotor and drive mechanism may now be simulated by an equivalent system composed of two bodies mounted on a weightless shaft so that the bodies may be vibrated torsionally with respect to one another. Figure 14 illustrates the equivalent system under study.



I_1 = 0.52601 lb.in.sec.^{2} (Equivalent crank and reciprocating masses)
I_2 = 0.01212 lb.in.sec.^{2} (Equivalent rotor and drive assembly)

Figure 14. Equivalent Torsional Vibration System.

Freberg and Kemler (97) state that the natural torsional frequency of such a system may be expressed by the following formula:

$$\text{Natural Frequency } (f) = \frac{1}{2\pi}\sqrt{\frac{\pi d^4 G(I_1 + I_2)}{32\ I_1\ I_2\ L}} \quad \text{(cycles per second)}$$

where the shear modulus of elasticity (G) is taken as equal to 11,500,000 lbs/in.^{2} (98). Substituting the appropriate values into the formula, we compute the natural frequency of the system for an assumed shaft diameter (d) of 5/8 inches to be 162.06 cycles per second. Similarly an assumed shaft diameter (d) of 3/4 inches yields a natural frequency of 233.37 cycles per second. Applying these frequencies to the formula (99):

$$\text{Critical Engine Speed} = \frac{60f}{N_o(n)} \quad \text{rpm}$$

where (n) refers to the number of power strokes in one revolution of the crankshaft (1/2 in this instance) and N_o refers to the order of excitation. We determine the following tabulation of critical engine speeds for the first six orders of excitation:

Table 2. Critical Engine Speeds.

Excitation Order No.	Critical Engine Speed 5/8 in. shaft diameter	Critical Engine Speed 3/4 in. shaft diameter
1	19,450 rpm	28,040 rpm
2	9,724 rpm	14,020 rpm
3	6,483 rpm	9,348 rpm
4	4,862 rpm	7,011 rpm
5	3,890 rpm	5,609 rpm
6	3,241 rpm	4,674 rpm

In view of the experimental nature of this design, it is desirable to design for at least equivalent speeds to contemporary engines with which competition is desired (See Table 7a). The use of a 3/4 inch rotor drive shaft makes it possible to operate within this range without danger of excitation by any of the first six orders. Speeds up to at least 9000 rpm will be safe from a torsional vibration standpoint if care is taken not to operate continuously at or near any of the critical engine speeds listed in Table 1 for the 4th, 5th or 6th orders.

Miscellaneous.--In concluding the design it will be necessary to house all the specified gear components in an oil tight gear box and cylinder head combination as indicated in Figure 7. This item is not considered critical from the engineering standpoint and an aluminum casting of generous proportions has been provided for the engine under design. Eight 5/16-inch, 20-thread high-grade steel bolts (100) are specified at 1-7/8 inch radius and 45° intervals about the cylinder head to provide an allowable maximum cylinder pressure of 1200 psi with a safety factor of 2.

The spark plug should be placed in a slightly recessed position approximately 1/2 inch from the bottom of the port when the valve is at the median position of its adjustment travel, and the position on the cylinder circumference should correspond to a port center position when the engine crank is oriented at 10° before top dead center on the compression stroke (101). A 10-millimeter number M-8 plug manufactured by the AC Spark Plug Company has been provided in this instance due to its compact dimensions.

As mentioned previously in the discussion of rotor balance, consideration of combustion cell compactness was necessary in order to provide the desired compression ratio range. From the preliminary investigation of allowable pressures, a maximum compression ratio of 15:1 was considered feasible. The engine dimensions as listed allow for a 1/4 inch piston travel, and this would provide a minimum compression ratio of 8.26:1. This range is believed to be adequate for all load demands and in line with present quality of "regular" gasolines. However, as a precaution, the compression range is intentionally lowered for preliminary test runs through use of the specified low dome Ford piston. The range to be obtained will be 6.78:1 to 11:1 using a 1/16 inch minimum piston clearance from the rotor skirt. Prolonged operation at high compressions should be guarded against due to the inherent design limitations of this piston; however, a number of firms exist which may be called upon for special requirements of this nature (102). After preliminary tests have been successfully negotiated, the ultimate 15:1 compression ratio may be obtained through use of a special piston providing a matching contour to the under surface of the rotor of 1/16 inch over-all.*

Carburetion is available from numerous standard sources; however, the products of Amal Carburetor, Ltd. of Birmingham, England are exceptionally desirable for single cylinder engines where large flow pulsations occur. A 1-3/8 inch throat diameter is specified.

Although it is not within the scope of this study to attack the problem of a "sensing" device which should activate the controller mechanism in such a way as to coordinate the optimum compression ratios with

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*Dimensions specified here are based on experimental measurements taken on the existing volume of the combustion cell and clearance volume between the Ford piston and rotor.

performance demands, sufficient information should be available from practical considerations and a study of the theoretical information presented in Appendix B to predict the output characteristics desired of such a device. It may be observed that the maximum available compression ratio (15:1) is theoretically desirable for all except relatively high manifold pressures while using fuels similar to the Cadillac test gasoline; however, smoothness of low speed operation is improved through utilization of minimum available compression. It is apparent that manifold pressure (or throttle setting) and engine speed may be integrated in the sensing device to provide the necessary information for desired control.

The final design is by no means considered complete at present; however, those problems which have been regarded as unique to this mechanism have been treated sufficiently to make feasible the fabrication of a laboratory test engine based on existing information and experience.

APPENDIX A

Recommendations for Tests and Development

Preliminaries.—Before any attempt is made to operate the assembled valve unit, certain precautions should be observed to protect the mechanism from unforseen difficulties and possible damage. The entire engine should be carefully inspected for specified clearances, giving special attention to smooth operation of the rotor and possible errors in the vertical adjustment of the cylinder which might cause interference between the rotor skirt and piston. The engine may be mounted in any conventional type laboratory test arrangement. "The Testing of High Speed Internal Combustion Engines" by Arthur W. Judge (103) is an excellent reference on the subject. It is preferable, as will be shown, that a direct current, series wound electrical dynamometer be used, so that "motoring" of the engine can also be accomplished. In addition to the subjects treated there for conventional type units, it is recommended that the engine be subjected to a lengthy "motoring" period during which frictional measurements are taken on the valve gear and plotted versus time. When the shape of this friction-time curve approaches zero, the tests may be commenced. It is suggested that a special piston be prepared for this phase since it is undesirable to remove the connecting rod from this type lower end unit. This piston head should be well vented by several large holes to relieve compression and rings should be removed to reduce friction. Since it is inconvenient to have separate systems, the valve gear and all other working parts may be supplied oil from the same source. It is important, however, that provision be made for adjustment of the supply pressure to the rotor control system. Adequate means must be provided for removing heat from the oil since a significant part of the cooling load will be handled by the lubricant circulated through the valve control gear. Any high grade motor oil of SAE 40 to 70 weight may be used for evaluation (104) although those recommended for aircraft or racing are suggested. Silicone oils appear promising due to their ability to operate at the high temperatures which may be expected near the rotor skirt. The Dow Corning Corporation has been very accomodating in correspondence relating to the subject and they have made available a supply of special light weight silicone lubricant which was developed through turbo-jet engine research. The designation of the fluid is Dow Corning F-4050 and the viscosity is equivalent to SAE 10 weight. (105) Although no authoritative information is available at this time, certain additive blends of molybdenum sulphate appear worthy of evaluation as galling deterrents. The use of conventional oils of this viscosity should not be attempted.

A transparent settling and inspection tank should be provided in the rotor oil return line and a filter unit in the engine supply line.

The compression control valve should be provided with a stop adjustment which will allow for the circulation of oil through the thrust chamber while operating at maximum compression ratios. It is further recommended that the cooling water be supplied around the circumference of the cylinder at the point of top dead center piston travel so that the flow may be divided into an upward component for the rotor and a downward component for the cylinder barrel. Coolant at both discharge points should be throttled to maintain exit temperatures between 180° and 200° Fahrenheit. It is preferable that high circulating rates be stressed rather than excessively cool entering temperature, since harmful components of the combustion products may be induced to condense on internal surfaces if coolant temperatures average less than approximately 160° F. (106)

Instrumentation.--Conventional instrumentation should be provided for evaluation purposes. In addition, a rotor oil supply pressure gage and discharge flow rate measurement device are desirable. A suitable reference system should be established so that the exact elevation of the rotor will be known at all times as well as that of the controller. Thermocouples (107) should be used generously around the rotor, especially in the vicinity of the skirt, the spark plug, the exhaust sealer and the exhaust sealer spring.

Initial Starting.--Methanol alcohol will allow the engine to operate at lower temperatures than any of the common commercial fuels (108), hence it is desirable for the initial "break-in" period while operation characteristics are undefined. Carburation must be adjusted accordingly. The spark plug should be disconnected while the engine is motored at high speed. (Approximately 2000 rpm is suggested.) When proper adjustments are complete so that the compression control mechanism responds properly without oscillation at all throttle settings and hydrodynamic lubrication is attained around the rotor, and no unusual temperatures develop from friction, the unit is ready for starting under its own power.

With spark set at top dead center, the compression adjusted for minimum pressures, and all facilities in operation, the throttle should be closed while the engine is again motored to some moderate speed of low vibration (approximately 1000 rpm). Slow opening of the throttle should cause firing to commence in the engine. Temperatures should be observed closely for abrupt rises and when conditions appear satisfactory, the engine may take over the task of motoring itself. The engine should be allowed a generous power "break-in" period before evaluation is attempted. Commercial upper cylinder lubricants may be added to the gasoline for extra protection.

Development.—The testing of the proposed design arrived at in Chapter III is likely to result in a number of developmental problems, especially of a mechanical nature. A few of the anticipated difficulties are:

(a) The provision of a better supply of oil to the rotor possibly featuring intermittent feed with surface scraper for collection of surplus lubricant.

(b) The reduction of rotor friction through decreased external dimensions and increased clearances.

(c) The adjustment of brush type port seals for minimum pressures required for good contact without chatter.

(d) The evaluation of various materials and pressure devices for brush seals. (Powdered metallurgy offers potentialities in materials such as the graphite impregnated Chrysler "oilite" bearing and the porous "Bost-Bronze" of Boston Gear Works. (109)

(e) The control of gas or oil leakage around the rotor possibly through fitting expansion rings in the labyrinth type grooves originally specified for the rotor. (Hastings, manufacturer number 121 gray iron piston compression rings are suggested.)

(f) The improvement of response in the control system to meet the sudden demands for lower compressions. (Ability to attain high compressions rapidly is of nebulous value.)

(g) The establishment of favorable factors in combustion cell and port design pursuant to modification of existing chamber.

(h) The optimization of port timing and dimensions for various operating requirements.

(i) The partial control of pumping losses at low manifold pressures through evaluation of various means of varying port widths during actual operation of the engine.

(j) The reduction of excess cylinder wear by means of hardening the specified mild steel barrel, or adoption of a conventional cast iron cylinder.

(k) The development of multiple cylinder engine applications for the rotary valve.

APPENDIX B.

Table 3. Predicted Engine Cycles for Varying Intake Pressures
R_c: $6\frac{1}{4}$, Mixture: 85% theoretical air

When: $P_1 = 4.0$ psia
Assumed: $f = 0.150$, $Temp_1 = 880$ °F; then $E_c = 1324.0$ B.T.U.

Point	Vol.(ft^3)	Temp(°F)	Press(psia)	H (B.T.U.)	E (B.T.U.)	S
1.	85	880	4.0	- - - -	75	- - - -
2.	13.6	1530	- - -	- - - -	1324	- - - -
3.	13.6	4870	150	- - - -	1554	0.6935
4.	85	3230	15.5	- - - -	990	0.6935
4!	90	3200	14.7	930	980	0.6935
5.	85	3200	14.7	879	926	0.6935
6.	13.6	3200	14.7	140	147	0.6936
6!	39.7	2420	4.0	97.5	113	0.6935

Corrections: $T_1 = 900$ °F, $V_1 = 86.8$ cu. ft., $f = 0.151$

When: $P_1 = 7.0$ psia
Assumed: $f = 0.078$, $Temp_1 = 675$ °F; then $E_c = 1411.4$ B.T.U.

Point	Vol.(ft^3)	Temp(°F)	Press(psia)	H (B.T.U.)	E (B.T.U.)	S
1	37.0	675	7.0	- - - -	32	- - - -
2	5.92	1215	- - -	- - - -	152	- - - -
3	5.92	4920	360	- - - -	1563.4	0.630
4	37.0	3240	37.0	- - - -	993	0.630
4'	77.0	2990	14.7	745	830	0.630
5	37.0	2690	14.7	368	399	0.630
6	5.92	2690	14.7	57.3	64.0	0.630
6'	10.8	2280	7.0	46.5	55.4	0.630

Corrections: $T_1 = 677$ °F, $V_1 = 37.1$ ft^3, $f = 0.077$

When: $P_1 = 9.0$ psia
Assumed: $f = 0.0595$, $Temp_1 = 642$ °F; then $E_c = 1433.5$ B.T.U.

Point	Vol.(ft^3)	Temp(°F)	Press(psia)	H (B.T.U.)	E (B.T.U.)	S
1	27.3	642	9.0	- - - -	26	- - - -
2	4.37	1170	- - -	- - - -	141	- - - -
3	4.37	4945	495	- - - -	1574.5	0.609
4	27.3	3260	33.0	- - - -	997	0.609
4'	73	2520	14.7	685	783	0.609
5	27.3	2520	14.7	256	292.5	0.609
6	4.37	2520	14.7	41.0	46.8	0.609
6'	108	2260	9.0	35.9	42.7	0.609

Corrections: $T_1 = 642$ °F, $V_1 = 27.3$ ft^3, $f = 0.0598$

(continued)

Table 3. Predicted Engine Cycles for Varying Intake Pressures (continued)

R_c: $6\frac{1}{4}$, Mixture: 85% theoretical air

When: P_1 = 10.0 psia
Assumed: f = 0.054, $Temp_1$ = 630 °F; E_c = 1441 B.T.U.

Point	Vol.(ft^3)	Temp(°F)	Press(psia)	H (B.T.U.)	E (B.T.U.)	S
1	24.0	630	10.0	----	23	----
2	3.84	1145	---	----	136	----
3	3.84	4960	555	----	1577	0.599
4	24.0	3265	57	----	1000	0.599
4'	71	2460	14.7	670	770	0.599
5	24.0	2460	14.7	227	260	0.599
6	3.84	2460	14.7	36.2	41.0	0.599
6'	5.25	2250	10.0	32.0	38.4	0.599

Corrections: T_1 = 630 °F, V_1 = 24.0 ft^3, f = 0.0542

When: P_1 = 11.0 psia
Assumed: f = 0.049, $Temp_1$ = 618 °F; E_c = 1446.7 B.T.U.

Point	Vol.(ft^3)	Temp(°F)	Press(psia)	H (B.T.U.)	E (B.T.U.)	S
1	21.45	618	11.0	----	20	----
2	3.43	1130	---	----	132	----
3	3.43	4970	620	----	1578.7	0.591
4	21.45	3270	66	----	1002	0.591
4'	68.5	2400	14.7	643	750	0.591
5	21.45	2400	14.7	201.5	235	0.591
6	3.43	2400	14.7	32.2	37.5	0.591
6'	88	2250	11.0	29.5	35.4	0.591

Corrections: T_1 = 620 °F, V_1 = 21.50 ft^3, f = 0.050

When: P_1 = 12.0 psia
Assumed: f = 0.046, $Temp_1$ = 612 °F, E_c = 1452.3 B.T.U.

Point	Vol.(ft^3)	Temp(°F)	Press(psia)	H (B.T.U.)	E (B.T.U.)	S
1	19.48	612	12.0	----	19	----
2	3.115	1115	---	----	128	----
3	3.115	4980	670	----	1580	0.5835
4	19.48	3275	71	----	1003	0.5835
4'	67.5	2375	14.7	629	738	0.5835
5	19.48	2375	14.7	181.5	213	0.5835
6	3.115	2375	14.7	29	34	0.5835
6'	80	2245	12.0	27.2	32.6	0.5835

Corrections: T_1 = 610 °F, V_1 = 19.43 ft^3, f = 0.046

(continued)

Table 3. Predicted Engine Cycles for Varying Intake Pressures (continued)
R_c: $6\frac{1}{4}$, Mixture: 85% theoretical air

When: $P_1 = 14.7$ psia
Assumed: $f = 0.039$, $Temp_1 = 600$ °F; $E_c = 1459$ B.T.U.

Point	Vol.(ft^3)	Temp(°F)	Press(psia)	H (B.T.U.)	E (B.T.U.)	S
1	15.5	600	14.7	----	16	----
2	2.48	1090	---	----	122	----
3	2.48	4990	850	----	1581	0.567
4	15.5	3260	86	----	998	0.567
4'	63.5	2225	14.7	589	707	0.567
5	15.5	2225	14.7	144	172.5	0.567
6	2.48	2225	14.7	23	27.6	0.567
6'	2.48	2225	14.7	23	27.6	0.567

Corrections: $T_1 = 600$ °F, $V_1 = 15.58$ ft^3, $f = 0.0390$

Table 4. Predicted Engine Cycles for Varying Intake Pressures
R_c: 7, Mixture: 85% theoretical air

When: $P_1 = 10.0$ psia
Assumed: $f = 0.0483$, $Temp_1 = 635$ °F, $E_c = 1447$ B.T.U.

Point	Vol.(ft^3)	Temp(°F)	Press(psia)	H (B.T.U.)	E (B.T.U.)	S
1	24.25	635	10	----	24	----
2	3.47	1195	---	----	147	----
3	3.47	5000	620	----	1594	0.5955
4	24.25	3200	54	----	980	0.5955
4'	70	2425	14.7	652	758	0.5955
5	24.25	2425	14.7	346.5	262	0.5955
6	24.25	2425	14.7	33.3	37.5	0.5955
6'	4.75	2230	10	28.9	34.8	0.5955

Corrections: $T_1 = 620$ °F, $V_1 = 23.70$ ft^3, $f = 0.0495$

Table 5. Predicted Engine Cycles for Varying Intake Pressures
R_c: 9 3/4, Mixture: 85% theoretical air

When: $P_1 = 4$ psia
Assumed: $f = 0.081$, $Temp_1 = 662$ °F; $E_c = 1409.3$ B.T.U.

Point	Vol.(ft^3)	Temp(°F)	Press(psia)	H (B.T.U.)	E (B.T.U.)	S
1	63.4	662	4.0	- - - -	30	- - - -
2	6.50	1370	- - -	- - - -	190	- - - -
3	6.50	4980	330	- - - -	1599	0.646
4	63.4	2980	19.2	- - - -	912	0.646
4'	80	2800	14.7	789	865	0.646
5	63.4	2800	14.7	625	686	0.646
6	6.50	2800	14.7	64.1	70.3	0.646
6'	18.7	2110	4.0	44.3	54.5	0.646

Corrections: $T_1 = 665$ °F, $V_1 = 63.7$ ft^3, $f = 0.0813$

When: $P_1 = 6.0$ psia
Assumed: $f = 0.061$, $Temp_1 = 631$ °F; $E_c = 1431.8$ B.T.U.

Point	Vol.(ft^3)	Temp(°F)	Press(psia)	H (B.T.U.)	E (B.T.U.)	S
1	38.9	631	6.0	- - - -	23	- - - -
2	3.99	1320	- - -	- - - -	177.5	- - - -
3	3.99	5020	530	- - - -	1609.3	0.608
4	38.9	2990	31	- - - -	915	0.608
4'	66.5	2520	14.7	684	783	0.608
5	38.9	2520	14.7	391	457	0.608
6	3.99	2520	14.7	41	47	0.608
6'	8.82	2070	6.0	31.8	39.6	0.608

Corrections: $T_1 = 632$ °F, $V_1 = 39.1$ ft^3, $f = 0.060$

When: $P_1 = 9.0$ psia
Assumed: $f = 0.040$, $Temp_1 = 595$ °F; $E_c = 1456.1$ B.T.U.

Point	Vol.(ft^3)	Temp(°F)	Press(psia)	H (B.T.U.)	E (B.T.U.)	S
1	25.25	595	9.0	- - - -	15	- - - -
2	2.59	1255	- - -	- - - -	162	- - - -
3	2.59	5070	820	- - - -	1618.1	0.577
4	25.25	2995	49	- - - -	920	0.577
4'	65.0	2310	14.7	612	725	0.577
5	25.25	2310	14.7	234	277	0.577
6	2.59	2310	14.7	24.5	29.0	0.577
6'	3.92	2080	9.0	21.2	26.4	0.577

Corrections: $T_1 = 593$ °F, $V_1 = 25.15$ ft^3, $f = 0.0398$

(continued)

Table 5. Predicted Engine Cycles for Varying Intake Pressures
(continued)
R_c: 9 3/4, Mixture: 85% theoretical air

When: $P_1 = 10.0$ psia
Assumed: $f = 0.036$, $Temp_1 = 587$ °F; $E_c = 1462$ B.T.U.

Point	Vol.(ft^3)	Temp(°F)	Press(psia)	H (B.T.U.)	E (B.T.U.)	S
1	22.4	587	10.0	- - - -	14	- - - -
2	2.30	1240	- - -	- - - -	158	- - - -
3	2.30	5083	920	- - - -	1462	0.569
4	22.4	3010	57	- - - -	922	0.569
4'	65.5	2250	14.7	592	710	0.569
5	22.4	2250	14.7	202	243	0.569
6	2.30	2250	14.7	21.6	25.9	0.569
6'	3.28	2066	10.0	19.3	24.4	0.569

Corrections: $T_1 = 587$ °F, $V_1 = 22.4$ ft^3, $f = 0.036$

When: $P_1 = 11$ psia
Assumed: $f = 0.033$, $Temp_1 = 580$ °F; $E_c = 1466$ B.T.U.

Point	Vol.(ft^3)	Temp(°F)	Press(psia)	H (B.T.U.)	E (B.T.U.)	S
1	20.1	580	11.0	- - - -	12.5	- - - -
2	2.06	1230	- - -	- - - -	155.5	- - - -
3	2.06	5092	1020	- - - -	1621.5	0.562
4	20.1	3015	62	- - - -	924	0.562
4'	63.8	2205	14.7	579	698	0.562
5	20.1	2205	14.7	183	220	0.562
6	2.06	2205	14.7	19.3	23.2	0.562
6'	2.69	2076	11.0	17.7	22.0	0.562

Corrections: $T_1 = 580$ °F, $V_1 = 20.1$ ft^3, $f = 0.0332$

When: $P_1 = 12$ psia
Assumed: $f = 0.031$, $Temp_1 = 575$ °F; $E_c = 1469.8$ B.T.U.

Point	Vol.(ft^3)	Temp(°F)	Press(psia)	H (B.T.U.)	E (B.T.U.)	S
1	18.2	575	12.0	- - - -	11	- - - -
2	1.87	1215	- - -	- - - -	152	- - - -
3	1.87	5100	1125	- - - -	1621.8	0.554
4	18.2	2010	67	- - - -	923	0.554
4'	62.5	2175	14.7	562	684	0.554
5	18.2	2175	14.7	164	199	0.554
6	1.87	2175	14.7	16.9	20.5	0.554
6'	2.19	2075	12.0	15.8	658	0.554

Corrections: $T_1 = 575$ °F, $V_1 = 18.2$ ft^3, $f = 0.030$

(continued)

Table 5. Predicted Engine Cycles for Varying Intake Pressures (continued)
R_c: 9 3/4, Mixture: 85% theoretical air

When: $P_1 = 14.7$ psia
Assumed:[1] $f = 0.025$, $Temp_1 = 565$ °F; $E_c = 1476$ B.T.U.

Point	Vol.(ft^3)	Temp(°F)	Press(psia)	H (B.T.U.)	E (B.T.U.)	S
1	14.6	565	14.7	----	9	----
2	1.50	1190	---	----	146	----
3	1.50	5115	1400	----	1622	----
4	14.6	3000	86	----	920	----
4'	59.5	2050	14.7	521	654	
5	14.6	2050	14.7	283	161	
6	1.50	2050	14.7	12.9	16.5	
6'	1.50	2050	14.7	13.1	16.5	

Corrections: $T_1 = 565$ °F, $V_1 = 14.6\ ft^3$, $f = 0.025$

Table 6. Predicted Engine Cycles for Varying Intake Pressures
R_c: 15, Mixture: 85% Theoretical Air

When: P_1 = 4.0 psia
Assumed: f = 0.054, $Temp_1$ = 615 °F; E_c = 1441 B.T.U.

Point	Vol.(ft^3)	Temp(°F)	Press(psia)	H (B.T.U.)	E (B.T.U.)	S
1	58.3	610	4	----	18	----
2	3.88	1450	---	----	210	----
3	3.88	5120	580	----	1651	0.6150
4	58.3	2750	19.8	----	847	0.6150
4'	73	2560	14.7	704	798	0.6150
5	58.3	2560	14.7	567	643	0.6150
6	3.98	2560	14.7	37.8	42.8	0.6150
6'	11.3	1950	4	27	34.1	0.6150

Corrections: T_1 = 610 °F, V_1 = 58.2 ft^3, f = 0.054

When: P_1 = 7.0 psia
Assumed: f = 0.033, $Temp_1$ = 580 °F; E_c = 1465 B.T.U.

Point	Vol.(ft^3)	Temp(°F)	Press(psia)	H (B.T.U.)	E (B.T.U.)	S
1	31.8	580	7	----	10	----
2	2.12	1395	---	----	95	----
3	2.12	5165	1070	----	1661	0.5695
4	31.8	2760	37	----	850	0.5695
4'	65.0	2250	14.7	592	710	0.5695
5	31.8	2250	14.7			0.5695
6	2.12	2250	14.7	19.2	23.1	0.5695
6'	3.80		7	15.7	20.1	0.5695

Corrections: T_1 = 585 °F, V_1 = 31.85 ft^3, f = 0.033

When: P_1 = 10.0 psia
Assumed: f = 0.024, $Temp_1$ = 560 °F; E_c = 1478 B.T.U.

Point	Vol.(ft^3)	Temp(°F)	Press(psia)	H (B.T.U.)	E (B.T.U.)	S
1	21.3	560	10	----	8	----
2	1.42	1355	---	----	186	----
3	1.42	5205	1580	----	1664	0.540
4	21.3	2775	56.5	----	850	0.540
4'	59.5	2080	14.7	532	661	0.540
5	21.3	2080	14.7	190.3	236.5	0.540
6	1.42	2080	14.7	12.7	15.8	0.540
6'	1.91	1915	10	480	616	0.540

Corrections: T_1 = 558 °F, V_1 = 21.3 ft^3, f = 0.024

(continued)

Table 6. Predicted Engine Cycles for Varying Intake Pressures (continued)
R_c: 15, Mixture: 85% Theoretical Air

When: $P_1 = 12.0$ psia
Assumed: $f = 0.020$, $Temp_1 = 550$ °F; $E_c = 1482$ B.T.U.

Point	Vol.(ft^3)	Temp(°F)	Press(psia)	H (B.T.U.)	E (B.T.U.)	S
1	17.52	552	12	- - - -	7	- - - -
2	1.17	1340	- - -	- - - -	182.5	- - - -
3	1.17	5230	1900	- - - -	1664.5	0.5240
4	17.52	2755	67	- - - -	848	0.5240
4'	57.5	1990	14.7	502	637	0.5240
5	17.52	1990	14.7			0.5240
6	1.17	1990	14.7	10.2	12.9	0.5240
6'	1.36	1910	12	9.5	12.4	0.5240

Corrections: $T_1 = 550$ °F, $V_1 = 17.48$ ft^3, $f = 0.020$

When: $P_1 = 14.7$ psia
Assumed: $f = 0.018$, $Temp_1 = 545$ °F; $E_c = 1485.3$ B.T.U.

Point	Vol.(ft^3)	Temp(°F)	Press(psia)	H (B.T.U.)	E (B.T.U.)	S
1	14.1	545	14.7	- - - -	5	- - - -
2	0.940	1315	- - -	- - - -	176	- - - -
3	0.940	5235	2350	- - - -	1661.3	0.506
4	14.1	2730	80	- - - -	841	0.506
4'	53.5	1890	14.7	470	608	0.506
5	14.1	1890	14.7	124	160	0.506
6	0.940	1890	14.7	8.3	10.7	0.506
6'	0.940	1890	14.7	8.26	10.7	0.506

Corrections: $T_1 = 545$ °F, $V_1 = 14.1$ ft^3, $f = 0.018$

Table 7a. Automobile Engine Specifications

Make, Model	Displ.	Max. HP at rpm	Max. Torque at rpm	Compr. Ratio	HP per in.^{3} displ.
American (1956 models).—					
Buick					
Series 50,60,70	322	255 @ 4400	341 @ 3200	9.50	0.793
Series 40	322	220 @ 4400	319 @ 2400	8.90	0.683
Cadillac					
Eldorado	365	305 @ 4700	400 @ 3200	9.75	0.836
Deville	365	285 @ 4600	400 @ 2800	9.75	0.782
Chevrolet					
V-8 (Standard)	265	205 @ 4600	268 @ 3000	9.25	0.774
Chrysler					
New Yorker	354	280 @ 4600	380 @ 2800	9.00	0.791
DeSoto					
Firedome	330	230 @ 4400	305 @ 2800	8.50	0.697
Fireflite	330	255 @ 4400	350 @ 3200	8.50	0.773
Ford					
V-8 (Standard)	272	173 @ 4400	260 @ 2400	8.00	0.636
Customline "Six"	223	137 @ 4200	202 @ 2200	8.00	0.615
Thunderbird (with overdrive)	312	225 @ 4600	324 @ 2600	9.00	0.722
Lincoln					
Capri & Premire	368	285 @ 4600	402 @ 3000	9.00	0.775
Mercury					
Monterey	312	210 @ 4600	312 @ 2600	8.00	0.673
Oldsmobile					
"88"	324.3	230 @ 4400	340 @ 2400	9.25	0.[illegible]
"Super 88" & "98"	324.3	240 @ 4400	350 @ 2800	9.25	0.739
Packard					
5688 (dual carbs.)	374	310 @ 4600	405 @ 2800	10.00	0.830
Plymouth					
V-8 (Standard)	277	187 @ 4400	265 @ 2400	8.00	0.675
Fury	303	240 @ 4800	310 @ 2800	9.25	0.793

(continued)

Table 7a. Automobile Engine Specifications (continued)

Make, Model	Displ.	Max. HP at rpm	Max. Torque at rpm	Compr. Ratio	HP per in.^{3} displ.
European.—					
Alfa Romeo					
1900	114.9	99 @ 5500	104 @ 3000	---	0.86
Ferrari					
375 Mexico	275.9	318 @ 6800	---	---	1.16
250 Mille Miglia	180.8	237 @ 7200	---	---	1.31
Fiat					
600 Saloon	38.6	21.5 @ 4600	28.9 @ 2800	7.00	0.56
Jaguar					
Type D	210	250 @ 6000	242 @ 4000	9.00	1.19
Lancia					
Aurelia	149.1	118 @ 5000	134 @ 3500	8.0	0.79
Mercedes					
SL 300	182.8	240 @ 6100	217 @ 4800	8.55	1.31
Nash-Frazer					
Targa Florio	120.3	105 @ 5000	123 @ 3750	8.50	0.87
Pegaso					
Z 102	150.5	165 @ 6500	138 @ 3900	---	1.10
Renault					
4-door saloon	45.6	21 @ 4100	33.25 @ 2000	7.25	0.46
Siata					
208 CS	121.8	120 @ 6300	107 @ 4200	---	0.99
Simca					
Aronde	74.5	45 @ 4500	61 @ 2600	6.80	0.60
Triumph					
TR 2	121.5	90 @ 4800	117 @ 3000	8.50	0.74
Experimental.—					
Cross Motorcycle	15	17.5 @ 6000			1.17
Aspin					
Aspin-Type German	30.4	Approx. 55 HP	---	9.00	Approx.1.8
Zimmerman	14.92	45 @ 12000	22 @ 6000(up)	12.00	3.0

Table 7b. Sample Aircraft Engine Specifications (1954).

Make, Model	Max. HP at rpm	Compr. Ratio	HP per cu.in.	Oil Consumpt. lb./HP hr.	Fuel Consumpt. lb./HP hr.	Fuel Req'd. (Octane)
Unsupercharged						
American.—						
Franklin 6V4	178@3000	7.0	0.53	0.010	0.51	80
Continental						
A-65	65@2300	6.3	0.38	0.009	0.50	73
C-85-12	85@2575	6.3	0.45	0.010	0.52	73
C-145	145@2700	7.0	0.48	0.017	0.51	80
O-470	225@2600	7.0	0.48	0.018	0.51	80
Jacobs R-755	300@2200	6.0	0.40	0.015	0.45	80
Lycoming O-235	115@2800	6.75	0.46	0.015	0.47	80
French.—						
S.N.E.C.M.A.						
Regnier	170@2500	7.25	0.40	0.018	0.48	100/130
Italian.—						
Alfa Romeo 115	225@2400	6.5	0.40	0.015	0.51	100/130
Spanish.—						
E.N.M.A. Flecha	90@2500	7.0	0.42	0.020	0.50	80/87
Supercharged (Boost Range: 43.9 to 80.8 in. Hg.)						
American.—						
Continental						
R-975	550@2400	6.3	0.56	0.015	0.50	115/145
Lycoming O-580	375@3300	7.3	0.65	0.015	0.45	91/96
Pratt & Whitney						
R-2800(CB)	2400@2800	6.75	0.86	0.015	0.42	100/130
R-4360(CB)	3500@2700	6.7	0.80	0.015	0.43	108/135
Wright						
(Turbo-Comp)						
18R-3350(D)	3500@2900	6.7	1.05	0.015	0.38	115/145
British.—						
Bristol*						
Hercules (738)	2040@2800	7.0	0.87	0.008	0.42	100/130
Centaurus (173)	2850@2800	7.2	0.93	0.008	0.42	100/130
Rolls Royce						
Griffon (57)	2455@2750	6.0	1.12	0.008	0.42	100/130
Russian.—						
ASH-90C	2200@2800	6.5	0.81	0.020	0.46	95

*Sleeve valve equipped.

Table 7c. Automobile Engine Valve Specifications.

	Intake		Exhaust		Valve	diam.	
Make, Model	Opens B.T.C.	Closes A.B.C.	Opens B.T.C.	Closes A.T.C.	Int. (in.)	Exh. (in.)	Lift (in.)
Buick							
Series 50,60,70	30°	82°	78°	44°	1.75	1.37	0.378
Series 40	25	77	75	42	1.75	1.37	0.378
Cadillac							
All models	39	105	81	63	1.75	1.56	0.451
DeSoto							
Firedome	4*	76	54	10	1.94	1.75	0.381 int.
Fireflite	15	57	49	15	1.94	1.75	0.375 exh.
Ford							
All V-8 mods.	12	54	58	8	1.78	1.51	0.385
"Six"	24	46	68	2	1.78	1.51	0.370
Lincoln							
Capri & Premiere	18	72	59	31	2.00	1.64	0.417
Mercury							
Monterey	12	54	58	8	1.75	1.51	0.360
Oldsmobile							
All models	$11\frac{1}{2}$	$52\frac{1}{2}$	51	13	1.75	1.56	0.418
Packard							
5688	14	62	54	18	2.00	1.69	0.398 int. 0.388 exh.
Mercedes							
SL300	11	53	37	11	---	---	---
Zimmerman							
Experimental (Rotary Valve)	20	70	70	20	---	---	N.A.**

*A.T.C. (after top center).

**Not applicable to rotary valve engines.

Table 8. Performance Characteristics and Their Relationship to Compression Ratio (R_c) and Intake Pressure (P_1)

R_c	P_1 (psia)	Pump Work in (B.T.U./lb.)	Net Work out (B.T.U./lb.)	MEP (psi)	Therm. Eff. (%)	Vol. Eff. (%)
6 1/4	4	127	282.0	21.4	21.30	15.6
	7	41.9	408.5	71.0	28.95	39.1
	9	23.6	438.9	103.5	30.65	53.9
	10	18.0	446.0	119.0	30.95	61.5
	11	12.1	452.6	135.5	31.30	69.2
	12	8.2	459.8	151.8	31.60	76.5
	14.7	0.	477.0	197.7	32.60	97.1
7	10	17.4	473.6	123.1	32.7	60.2
9 3/4	4	106.0	421.0	40.0	29.9	21.2
	6	54.3	485.5	75.1	33.9	35.4
	9	25.9	525.2	120.0	36.1	55.8
	10	17.8	536.2	144.3	36.7	62.6
	11	12.5	542.0	163.0	37.0	70.6
	12	8.3	549.5	181.0	37.4	77.8
	14.7	0.0	565.0	231.0	38.3	97.2
15	4	104.4	507.6	50.3	35.25	22.9
	7	41.4	583.6	107.0	39.9	43.1
	10	17.2	618.8	168.0	41.9	65.5
	12	9.6	631.4	209.3	42.65	79.0
	14.7	0.0	649.3	266.0	43.7	97.8

Table 9. Test Data 1948 Six-Cylinder Chevrolet Engine

Maximum Rated Horsepower..............92
Barometer..............................28.05 in Hg. = 14.27 psia.
Wet Bulb Temperature..................59°F
Dry Bulb Temperature..................69°F
Volume 1 lb. dry air..................13.05 cu. ft.
Compression Ratio.....................6.25:1
Cylinder Bore..........................3.5 in.
Piston Stroke..........................3.75 in.
Heating Value of Fuel.................19,500 B.T.U./lb.
Engine Speed...........................2600 r.p.m.
Spark Advance..........................33°B.T.C.
Displacement...........................232 cu. in.

Run No.	Fuel-Air Ratio	Brake Horsepower	IMEP (calculated)	Volumetric Efficiency	Thermal Efficiency
1	0.062	7.9	13.4	21.7	10.8
2	0.064	19.5	33.1	30.2	18.5
3	0.062	32.4	55.0	41.3	23.2
4	0.060	46.5	79.0	52.6	26.8
5	0.062	60.6	103	64.6	27.2

---Manifold pressure was varied---

Note that volumetric efficiency is approximated through the use of a 1-1/2 in. diameter thin plate orifice on the intake duct (110).

$$\text{BMEP} = \frac{\text{B.H.P.}\ (10^7)}{9.917\ N\ L\ n\ D^2}$$

$$\eta = 0.98\left(1 - \frac{0.3}{D}\right) \quad (29)\ (\text{Estimated});\quad \eta = 89.5\%$$

$$\text{IMEP} = \frac{\text{BMEP}}{\eta}$$

APPENDIX C

Design of Reed Check-Valve:

From Figure 15 it can be seen that the reed valve can be reduced to simulate a centilever with concentrated loading.

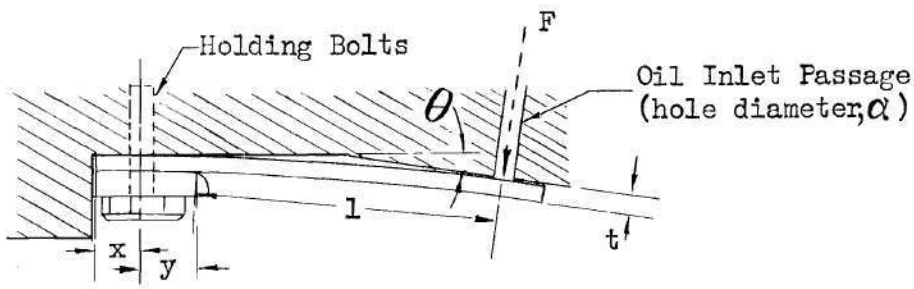


Figure 15.

Reed Check and Back-Pressure Valve (R, Figure 7)

Specifications for the valve may be computed from arbitrary differential back pressure of 50 psi from the deflection formula:

$$\delta = \frac{Fl^3}{3EI} \quad (111)$$

where: δ is reed deflection
F is total force acting on the reed from differential oil pressure (consider pressure to act only over area of oil supply hole)
l is effective length of the cantilever
E is Young's modulus of elasticity for reed material
I is moment of inertia of reed cross section

Chromium - Vanadium steel, SAE 6152 quenched and drawn at 1150°F (112) is specified as material for the reed. It is determined that the reed should have the following dimensions as indicated in Figure 15.

length overall..............1 5/8 inch.
length (l)..................1 inch.
width (w)...................0.5 inch.
thickness (t)...............0.010 inch (Brown and Sharp gage 30) (113)
dimension (x)...............0.25 inch.
dimension (y)...............0.125 inch.
hole diameter (α)..........0.125 inch.

Angle θ should be machined 14.1 degrees in order to cause proper seating of the valve at 50° psi oil pressure. The reed may be attached by two 3/32 inch diameter steel bolts acting on a rounded face clamp as indicated. The valve is recessed into the upper surface of the chamber near the cylinder wall to allow clearance for operation of the rotor. Stress levels are low in the reed (in the order of 15,000 psi operating under 100 psig pressure O_1). Due to the oil environment, damping is considered adequate to eliminate consideration of critical vibrations from engine excitation.

As an added control precaution, a needle adjustment valve should be fitted in the oil supply line to aid in the establishment of desirable supply rate for control of valve position and oil discharge temperature.

APPENDIX D

DYNAMIC BALANCE OF ROTOR

Center of Gravity of Removed Mass.--In order to compute the correct compensations for a dynamic balance of the rotor, it is necessary to know the exact weight (W_t) and gravity center (CG_r) of the metal removed in the machining of the combustion cell. This information was obtained experimentally by measuring the weight of the rotor and locating its longitudinal balance point before and after the machining of the port; then determining the torsional correction necessary to bring about a static balance. The procedures used for these measurements are indicated in Figures 16 and 17.

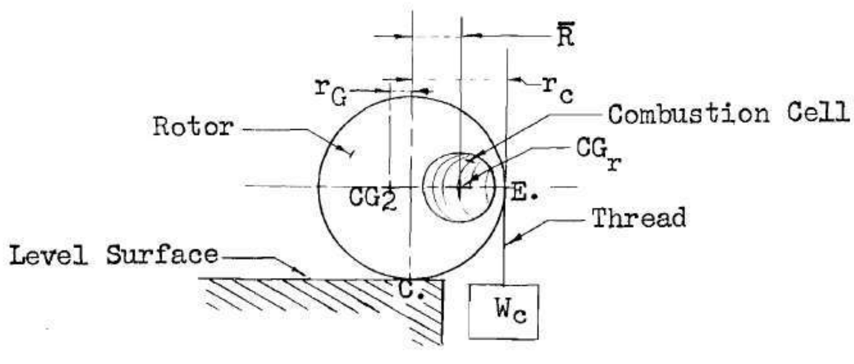
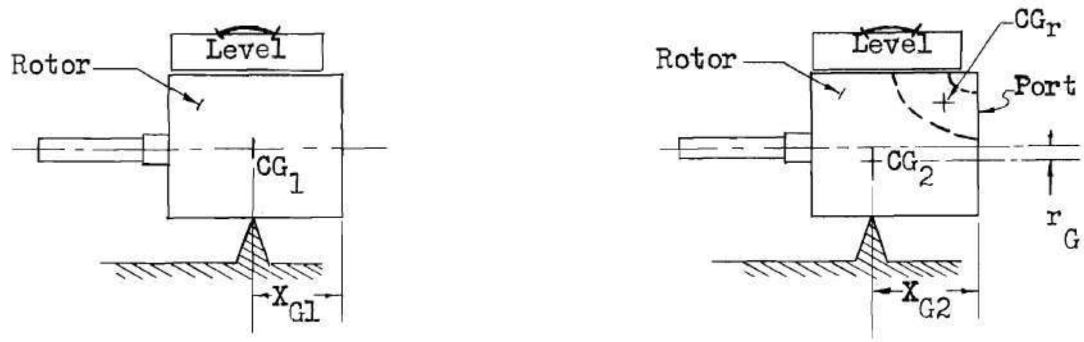


Figure 16.

Experimental Determination of Radial Unbalance of Rotor.

Compensation weight (W_c) is adjusted until the system is in equilibrium while resting on point C. Weight (W_c) may be considered concentrated at point E, hence the torsion required to balance the rotor as indicated is the product of the moment arm (r_c) and weight (W_c). This value was computed:

$$W_c \times r_c = 6.67 \text{ inch ounces.}$$



Before Machining Port After Machining Port

Figure 17.

Experimental Determination of Longitudinal Balance Points of a Rotor.

True horizontal is indicated by a hand level suspended in the near proximity of the upper rotor surface and appropriate adjustments are made to balance the rotor in a level attitude on the knife edge. The center of gravity must lie vertically over the balance point.

The following values were measured:

Rotor weight without port (W_{tr})........................6.957 lbs.
Balance dimension X_{G1}..................................1.993 in.
Rotor weight with port (W_t)..............................6.428 lbs.
Balance dimension X_{G2}..................................2.055 in.
Weight of material removed in port (W_r)..................0.429 lbs.

Letting $\overline{H}$ represent the axial distance from the rotor skirt to the center of gravity of removed combustion cell mass (CG_r) and $\overline{R}$ the radial distance of CG_r from the rotor axis, we evaluate the information obtained from the experiments outlined in Figures 16 and 17.

Longitudinal Balance, Figure 17.

Prior to machining rotor port, $\sum \text{Moments}_{CG_1} = 0$

After machining rotor port, $\sum \text{Moments}_{CG_1} = (X_{G2} - X_{G1})W_t$

Hence, $\sum \text{Moments}_{CG_1} = 0.062$ in. x 6.428 lb. = 0.398 lb.in.

Let p = Axial distance from CG_1 to CG_r

then, 0.398 lb.in. = p W_r = p x 0.429 lb.in.

and p = 0.929 inches.

Therefore, $\overline{H} = X_{G1} - p = 1.993 - 0.929$

and $\overline{H} = 1.064$ in.

Lateral Balance, Figure 16.

Prior to machining rotor port, $\sum \text{Moments}_c = 0.$

After machining rotor port, $\sum \text{Moments}_c = 6.67$ inch ounces.

Since $W_r = 0.429$ lbs. = 6.87 ounces

$\overline{R} = 0.971$ inches.

Analytical Prediction of Compensations Required for Dyanmic Balance.

Numerous arrangements are possible for providing a dynamic balance of the rotor; however, in view of inherent heat dissipation advantages, non-disturbance of rotor geometry and simplicity, the arrangement indicated in Figure 18 is recommended.

Since all rotor plugs are selected of the same length (a=2.625 in.), the plug center of gravity will lie in a plane 2.088 inches above and parallel to the rotor skirt. The conditions for static and dynamic balance are verified here:

Static Equation: $$\sum^{n} W_n r_n = 0 \qquad (114)$$

$$Wr\,\overline{R} - W_a\,(2\,r_a) - W_b(2\,r_p + r_c) + W_f r_f = 0.$$

density of: (91) Steel - 4.48 ounces per cu. inch.
Aluminum - 1.53 ounces per cu. inch.
Cast iron - 4.10 ounces per cu. inch.
Copper - 5.15 ounces per cu. inch.
Lead - 6.57 ounces per cu. inch.

Then weights: W_n = Volume material x density change

$$W_b = W_c = \frac{\pi}{4}(D_a^{\,2})(\text{Length}_a)(\text{Density Steel} - \text{Density Aluminum})$$

$$W_b = \frac{\pi}{4}(0.8025)^2(2.625)(4.48 - 1.53) \text{ ounces}$$

$$W_b = W_c = 3.92 \text{ ounces}$$

Similarly,

$$W_a = 0.945 \text{ ounces}$$

$$W_f = 1.57 \text{ ounces}$$

Substituting into static equation,

$$6.87(0.971) - 0.945(2x0.780) - 3.92(2x0.500+0.930)+1.57(1.50) = 0.$$

(check)

Dynamic Equation: $\sum^{n} W_n\, r_n\, X_n = 0.$ (115)

$$W_r\ \bar{R}\ \bar{H} - 2W_a\, r_a\, a - 2W_b\, r_b\, a - W_c\, r_c\, a + W_f\, r_f\, b = 0$$

Substituting,

$$6.87(0.971)1.064 - 2(0.945)0.780(2.088) - 2(3.92)0.500(2.088) - 3.92(0.930)2.088 \quad 1.57(1.50)5.00 = 0$$

(check)

Such computations for balance may be very precise; however, overall accuracy is limited by measurement techniques and the specified compensations should be verified or improved experimentally by a conventional dynamic balancing machine.

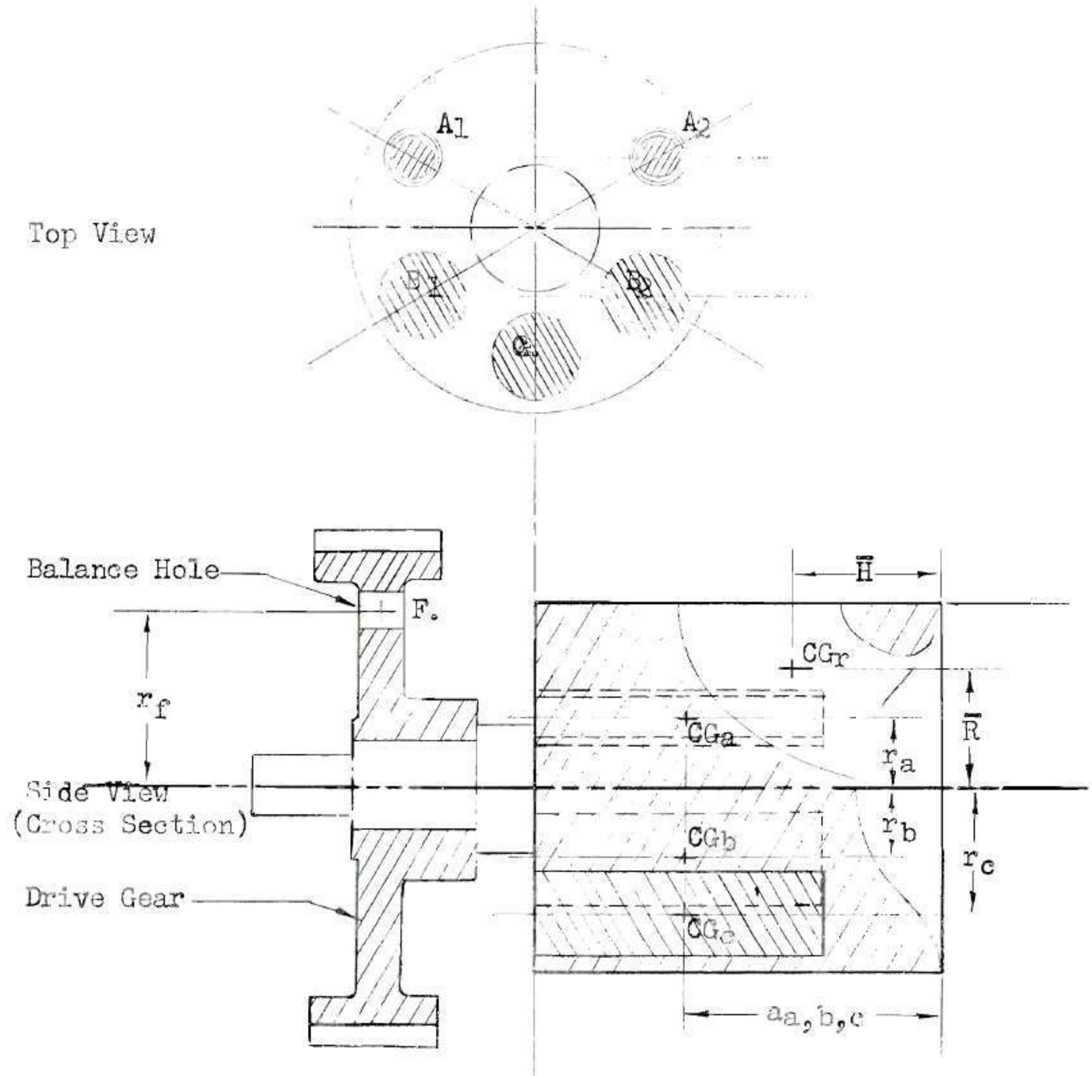


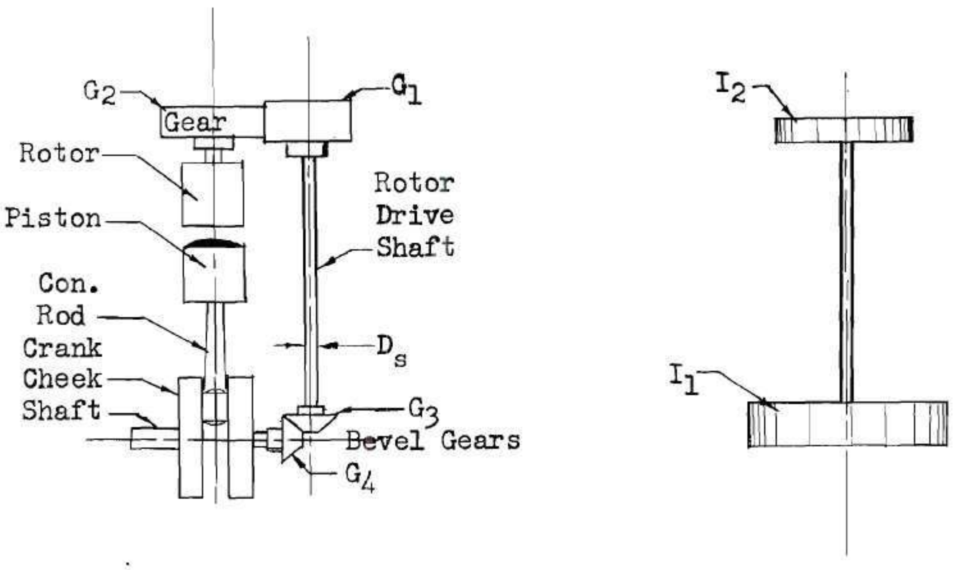
Figure 18. Dynamic Rotor Balancing System.

Specifications:
Plugs A_1 and A_2 (Lead centered copper)
length, 2-5/8 in.; diameter, 5/8 in. copper, 3/8 in. lead;
r_a = 0.780 in.
Plugs B_1, B_2, and C (Aluminum)
length, 2-5/8 in.; diameter, 0.[illegible]025 in.
r_b = 0.500 in.; r_c = 0.930 in.
Drilled Cavity F in Drive Gear (Cast iron)
length, 0.425 in.; diameter, 1.063 in.
r_f = 1.5 in.

APPENDIX E

TORSIONAL VIBRATION ANALYSIS OF ROTOR DRIVE SYSTEM

Rotor Drive Details.—The speed of the engine crank is used as reference and the summation of moments of inertia for all components of the valve and upper drive mechanism are corrected to that speed in order to obtain an "equivalent" mass which is considered to vibrate torsionally through the rotor drive shaft with the summed inertias of the crank and reciprocating parts of the engine "lower end."



Actual system
(a)

Equivalent system
(b)

Figure 19.

Schematic Representation of Significant Rotor Drive Details

Due to space limitations and strength demands, the following standard gears were selected by manufacturers' part number from the

Boston Gear Works catalog of 1955 for use as indicated in Figure 19(a):

Spur drive, G_1- - - - - - - - - Boston NF24B, 10 pitch, 2.400 in. pitch diameter, steel.
Spur drive, G_2- - - - - - - - - Boston NF48, 10 pitch, 4.800 in. pitch diameter, gray iron.
Bevel drive, G_3 and G_4- - - - Boston L129Y, 10 pitch, 45 degree miter, steel.

Spur drive gear, G_2, is machined further to meet dimension requirements indicated in Figure 12. The width of this gear is reduced by 0.250 inches to allow adjustment tolerance for the rotor while retaining full tooth contact with gear, G_1. The finished appearance of gear, G_2 and the rotor is presented in Figure 13.

Application of the Bifilar Pendulum in Inertia Measurement.--The rotor and drive gear (G_2) assembly was suspended in a conventional bifilar pendulum arrangement(93) and the natural period of oscillation measured at 1.002 seconds per cycle. This value was substituted into the bifilar formula:

$$\bar{I} = \frac{W_r{}^2 \tau^2}{4\pi^2 l} \quad (93)$$

where: Weight of assembly and holder (W)- - - - 8.518 lbs.
Radius of pendulum filament oscillation (4)---1.0025 in.
Period of oscillation (τ)- - - - - - - - 1.002 sec/cycle.
Length of filaments (l) - - - - - - - - - 23.30 in.

The moment of inertia for the rotor assembly and pendulum holder is then computed to be 0.03678 lb.-in.-sec.^{2}

The inertia of the holder was similarly measured as 0.000940 lb. in.sec.^{2}; hence the inertia of the rotor and gear assembly alone, ($I_{r,s2}$), is calculated to be 0.03584 lb.in.sec^2.

<u>Application of the Knife-Edge Pendulum in Inertia Measurement</u>.—The spur drive, G_1, and bevel gears, G_3 and G_4, were suspended along their inner shaft hole surfaces on a knife edge so as to form pendulums (94). The natural oscillation periods of the knife edge pendulums were observed and inertia was computed about the gear center-line axis by formula:

$$\bar{I} = \frac{Wr\tau^2}{4\pi^2} - r^2 \frac{W}{g} \qquad (94, 116)$$

Moment of inertia of spur gear, G_1, is computed where:

Weight of gear (W)- - - - - - - - - - - - - - -1.740 lbs.
Radius of shaft hole (r)- - - - - - - - - - - -0.3750 in.
Period of natural oscillation (τ)- - - - - 0.4790 sec.per. cycle

Then, $I_{s1} = 0.003158$ lb.in.sec.^{2}.

Similarly, the moment of inertia of bevel gears, G_3 and G_4, are computed where:

W - - - - - - - 0.730 lbs.
r - - - - - - - 0.3125 in.
- - - - - - - 0.4027 sec.per cycle

Then, $I_{b3} = I_{b4} = 0.001437$ lb. in. sec.^{2}

<u>Equivalent Moment of Inertia (I_{e1}), Rotor and Drive, Top End</u>.—Since the rotor and gear, G_2, travel at half engine speed, the equivalent inertia must be corrected to reference speed. The following formula then applies:

$$I_{e1} = I_{s1} + (1/2)^2 I_{r,s2} \qquad (117)$$

Specifically, $I_{e1} = 0.012118$ lb. in. sec.^{2}

Equivalent Moment of Inertia (I_{e2}), Crank End.—The crank end of the engine is considered to include the crank journal, Two flywheel cheeks, main reciprocating parts, shaft ends and bevel gears, G_3 and G_4. The flywheels are divided into small volumes and the inertia of each segment is computed with respect to the rotational axis of the crank shaft. Formulas based on the simplest geometries are employed (118). The inertia of respective components indicated in Figure 20 are computed to be:

$I_a = 0.02570$ lb. in. sec.^{2}

$I_b = 0.01097$ lb. in. sec.^{2}

$I_c = 0.005075$ lb. in. sec.^{2}

$I_d = 0.001191$ lb. in. sec.^{2}

The crank journal is approximately 1.125 inches long and 1.500 inches diameter. Since the engine stroke is 3.78 inches, the inertia of the journal is computed to be

$I_j = 0.005423$ lb. in. sec.^{2}

The equivalent inertia of the piston assembly and connecting rod is approximated by formula:

$$I_{er} = \left[\frac{W_{r,c}}{g} + \frac{W_{r,p} + W_p}{2g} \right] r_s^{\,2} \qquad (117)$$

where, weight of rod concentrated at the piston ($W_{r,p}$) is assumed to be 1/3 of total rod weight (W_r)
and remaining 2/3 weight of the rod ($W_{r,c}$) is assumed concentrated at the crank journal.

Weight of piston assembly (W_p) - - - - - - - - -	1.040 lbs.
$W_{r,p}$ -	0.550 lbs.
$W_{r,c}$ -	1.100 lbs.
Radius of engine stroke (r_s) - - - - - - - - - -	1.890 in.

The equivalent inertia of reciprocating parts is then computed to be:

$I_{er} = 0.017537$ lb. in. sec.^{2}

The shaft ends are considered equivalent to two segments of length, 3 in. and diameter, 0.500 in., and of length, 3.25 in. and diameter, 1.00 inches. By computation the total inertia of shaft ends (I_{sh}) is:

$$I_{sh} = 0.000265 \text{ lb. in. sec.}^2$$

With attention to the geometric division of the crank checks indicated in Figure 20 the equivalent inertia (I_{e2}) is estimated from a summation of component parts of the lower end:

$$I_{e2} = 2\left[I_b - \tfrac{1}{2}I_a - I_c - I_d\right] + I_j + I_{sh} + I_{er}$$

$$I_{e2} = 0.52595 \text{ lb. in. sec.}^2$$

Natural Torsional Vibration Frequencies.—Since the equivalent inertias, I_{e1} and I_{e2}, have been computed, it is desirable to select a coupling drive shaft diameter (D_s) which will be stiff enough to avoid a low natural frequency of torsional vibration which is likely to be excited by one of the lower orders engine firing pulsations. The natural frequency for a given shaft diameter, D_s, is computed by the formula:

$$\text{Natural Frequency } (f_n) = \sqrt{\frac{\pi D_s^4 G(I_{e1} + I_{e2})}{32 I_{e1} I_{e2} L}} \qquad (119)$$

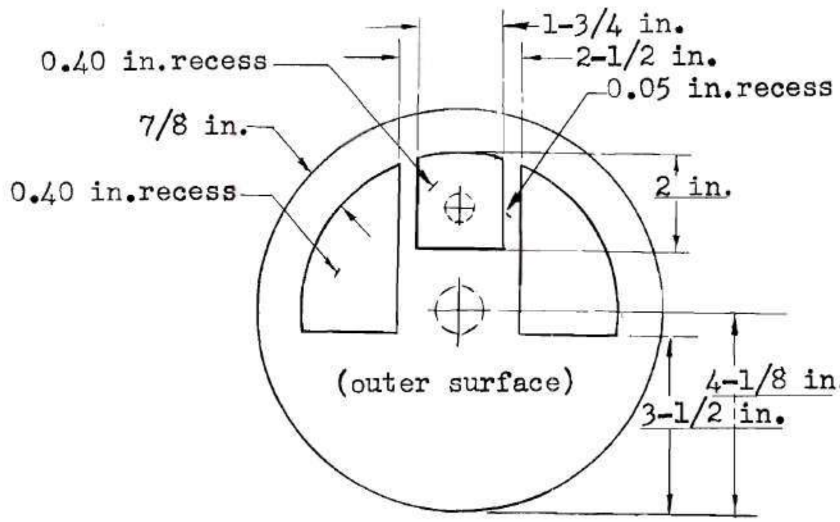
where the shear modulus of the steel shaft is taken as $G =$ 11,500,000 psi. and the fixed length of the shaft L is 14 inches.

Natural frequencies are computed for D_s; 5/8 inch and D_s, 3/4 inch assuming the weight of the shaft to have negligible influence on the frequencies.

These calculated frequencies are:

f_n = 9,724 cycles per minute*, (D_s 5/8 inch)

f_n = 14,020 cycles per minute*, (D_s 3/4 inch)



Crank Cheek

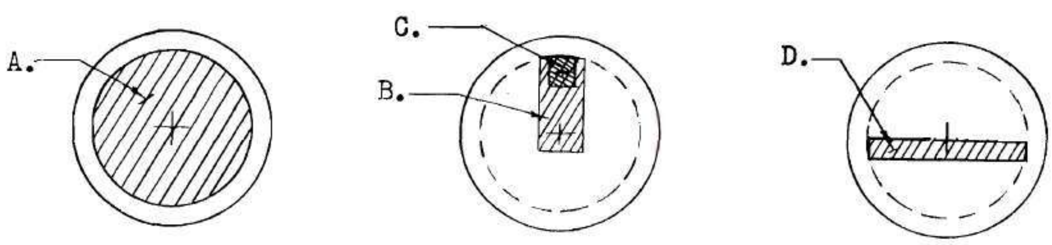


Figure 20.

Segmentation of Crank Cheek for Inertia Analysis

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*Computations have been carried to a large number of significant places in the interest of accuracy. Although the values stated here are rounded to four significant figures, results are probably no better than 5 per cent accurate in view of necessary assumptions.

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